

Proceedings of:

The Eighteenth Annual Meeting

The American Society for Precision Engineering

October 26-31, 2003

DoubleTree Jantzen Beach Hotel

Portland, Oregon

The American Society for Precision Engineering (ASPE) is a multidisciplinary professional and technical society concerned with research and development, design, manufacture and measurement of high accuracy components and systems. ASPE activities encompass relevant aspects of mechanical, electronic, optical and production engineering, physics, chemistry, and computer and materials science. Membership is open to anyone interested in any aspect of precision engineering.

Founded in 1986, ASPE provides a new focus for a diverse but important community. Other professional organizations have covered aspects of precision engineering, always as a sideline to their principal goals. ASPE is based on the core of generic concepts necessary to achieve precision in any application; independent of discipline, ASPE intends to be the focus for precision technology — and to represent all facets from research to application.

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Preface

This book comprises the proceedings of the 2003 Annual Meeting. The contributions reflect the authors' opinions and are published as presented to ASPE, without change. Their inclusion in this publication does not necessarily constitute endorsement by the ASPE, or its editorial staff.

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2003

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Welcome Note

Welcome to Portland for the Eighteenth Annual Meeting of The American Society for Precision Engineering, co-sponsored by JSPE. The growth and success of the Society reflects the increasing importance of precision engineering in a wide variety of fields of endeavor, from manufacturing to microelectronics to basic science. The Society serves as a focus for precision engineers across all of these fields. The Annual Meeting has evolved into a premier international forum for the exchange of ideas and presentation of research results relating to precision engineering, metrology, controls, and system integration. Precision engineers and scientists from private industry, government laboratories, and universities meet to learn about the latest developments and to exchange ideas about the future directions of these technologies.

This year's meeting continues the tradition of offering two full days of tutorials—including 12 new tutorials—presented by some of the foremost precision engineers and scientists in the world. The technical sessions and poster presentations offer the latest research results in the areas of ultraprecision machining, metrology, controls, precision grinding, micro-positioning, material processing, design, precision transducers, surface profilometry, machine tool metrology, and error compensation. The commercial exhibit will provide attendees with the opportunity to view and discuss the latest precision engineering equipment, products, and services that are commercially available. The open forum will give attendees the opportunity to express their opinions and pose questions for discussion by the group, enhancing the exchange of information.

The conference organizing committee is proud to present the program for the Eighteenth Annual Meeting of the ASPE. We welcome your participation and your feedback on how to make next year's meeting even better.

2003 Organizing & Technical Program Committee

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Mr. Craig Richard Forest, Massachusetts Institute of Technology

R.V. Jones Memorial Scholarship

2003 Annual Meeting

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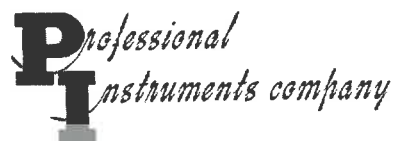
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Program

2003 ASPE Annual Meeting

<i>Time</i>	<i>Sun., Oct. 26</i>	<i>Mon., Oct. 27</i>	<i>Tues., Oct. 28</i>	<i>Wed., Oct. 29</i>	<i>Thurs., Oct. 30</i>	<i>Fri., Oct. 31</i>
8:00	Registration	Registration	Registration			
9:00	Tutorials	Tutorials	Technical Session I	Technical Session III	Technical Session VI	Technical Tours
10:00			Break	Break	Break	
11:00			Technical Session II	Technical Session IV	Technical Session VII	
12:00						
1:00			Lunch and Committee Meetings	Awards Lunch and Business Meeting	Lunch and Roundtable Discussions	
2:00	Tutorials	Tutorials	Commercial Session	Poster Session	Technical Session VIII	
3:00			Break	Break		
4:00			Poster Session	Technical Session V		
5:00			Hospitality Hour	Hospitality Hour		
6:00				Open Forum		
7:00		Keynote Address				
8:00		Welcome Reception	Off-site Dinner Banquet			
9:00						

Business Forum

Monday, October 27, 8:00 a.m. - 4:00 p.m.

*Session Co-Chairs: William J. Bryan, 3M Company and
Andrew J. Devitt, New Way Machine Components*

The ASPE Business Forum is open to all attendees of the annual meeting and will be of particular benefit to those with interests in business or the management of technology. The program will combine business and technology issues pertinent to leaders in the precision engineering community. It will be held this year on Monday October 27, 2003, just prior to the start of the formal ASPE meeting and overlapping one day with the Tutorial Program. The program will consist of morning and afternoon talks, a lunch break, and ample time to network with speakers and other attendees. The forum should conclude by mid-afternoon. Sustaining Corporate Sponsors of the ASPE are welcome to send one attendee to the Business Forum free of charge. Additional participants from the same Sustaining Corporate Sponsor, Corporate Sponsors, or from other organizations will be charged a fee of \$250.

Keynote Address

Milton Chang

Monday, October 27, 6:30 p.m.

Session Chair: Michele H. Miller, Michigan Technological University



"Confessions of a Technical Entrepreneur"

Dr. Milton Chang will share his insights on entrepreneurship which he learned the hard way over thirty years. Starting off as a research engineer and having been on both sides of the fence as an entrepreneur and as an investor, he will present a balanced view on how to go about learning, preparing, starting, and managing a business as well as working effectively with venture capital investors. He is a strong believer that entrepreneurship can be a satisfying and productive career for engineers, and will look back and reflect on his experience as a student, an engineer, a businessman, an angel investor, and a venture capitalist.

Milton Chang is Managing Director of Incubic (www.incubic.com), a venture fund in Silicon Valley "to help entrepreneurs build great companies." He earned a BS with Highest Honors from the University of Illinois, and MS and Ph.D. degrees from the California Institute of Technology, all in Electrical Engineering. Milton worked briefly as a research engineer before joining a startup company to begin his entrepreneurial career. He was president/CEO of Newport Corporation and New Focus, and has incubated more than a dozen companies with nearly a perfect record. He is currently on the boards of Arcturus Engineering, Lightwave Electronics, OEpic, OpVista, Rockwell Scientific, and YesVideo. He is active in the technical community and has received a number of prestigious awards. He was recently President of the IEEE Laser Electro-Optical Society, and writes monthly business columns for Laser Focus World and Photonics Spectra.

Commercial Session

Tuesday, October 28, 1:30 - 3:00 p.m.

Session Chairman: Alex Sohn, North Carolina State University

The Commercial Session will take place early in the week on Tuesday afternoon. This is a special session where company representatives are invited to make brief presentations (five minutes) on new products associated with precision engineering. This provides participants with the opportunity to receive timely information on new technologies that have been commercialized into products and services, as well as information on advances that can be expected in the near future.

Open Forum

Wednesday, October 29, 5:30 - 6:30 p.m.

Session Chairman: Eric R. Marsh (The Pennsylvania State University)

The Open Forum was created to encourage dissemination of new and significant results, as well as observations on precision engineering subjects and other topics of interest to the Society. So bring your overheads, problems, and experiences to this enlightening hour of impromptu discussions! Signup sheets will be available at the conference registration table.

Technical and Poster Sessions Index

The technical program for the 2003 ASPE Annual Meeting contains over 150 papers on precision engineering advances. Papers that lent themselves to significant verbal interaction, concise but important discoveries, or strongly visual or tactile subjects have been selected for the poster session. Authors will be present to discuss their work on Tuesday, October 28, from 3:30 to 5:00 p.m., and again on Wednesday, October 29, from 1:30 to 3:00 p.m.

Session I

Micro Manufacturing

Tuesday, October 28, 2003, 8:15 AM - 10:00 AM

Session Chair: E. Clayton Teague (National Science and Technology Council/NIST)

1. **Achieving Precision Through System Engineering (Invited Paper)**
van Eijk, J. (Delft University of Technology/Philips CFT); Soemers, H. M. J. R. (Twente University of Technology/Philips CFT); and Franse, J. (Philips CFT) page 13
2. **Dimensional Challenges in Printhead Micromachining and Inkjet Writing Systems (Invited Paper)***
Saul, K. (Hewlett Packard Corporation)
3. **Microtechnology-based Energy and Chemical Systems and Multi-scale Fabrication (Invited Paper)**
Paul, B. K.; Drost, K. M. (Oregon State University) page 19

Session II

Optical Systems and Metrology

Tuesday, October 28, 2003, 10:30 AM - 12:00 PM

Session Chair: Vivek G. Badami (Corning Tropel Corporation)

1. **High-NA Interference Microscopy of Complex Surface Structures (Invited Paper)**
de Groot, P. J.; Colonna de Lega, X.; Grigg, D. A. (Zygo Corporation) page 23
2. **Interferometric Thickness Measurement of Free Form Silicon Wafers**
Jansen, M.; Haitjema, H.; Schellekens, P. H. J. (Eindhoven University of Technology) page 27
3. **Roundness, Angle, Straightness and Waviness Measurements on Recessed Cones Using Scanning White-light Interferometry**
Colonna de Lega, X.; de Groot, P. J. (Zygo Corporation) page 31
4. **Stabilization for Atomic Tracking Control of the Scanning Tunneling Microscope Tip by Referring Regular Crystalline Surface**
Aketagawa, M.; Rerkkumsup, P.; Takada, K.; Togawa, Y.; Thinh, N. T.; Kozuma, Y. (Nagaoka University of Technology) page 35

* Abstract unavailable at press time

Session III

Equipment Design

Wednesday, October 29, 2003, 8:30 AM - 10:00 AM

Session Chair: Anthony E. Gee (University College, London)

1. **Lens Mounting Technology for Precision Lithography Optics (Invited Paper)***
Watson, D. (Nikon Research Corporation of America)
2. **Thin Glass Optic and Silicon Wafer Deformation and Kinematic Constraint**
Forest, C. R.; Akilian, M.; Vincent, G.; Lamure, A.; Schattenburg, M. L. (Massachusetts Institute of Technology) page 39
3. **Dynamic Friction Coefficient Measurements: Device and Uncertainty Analysis**
Schmitz, T. L.; Action, J.; Ziegert, J.C.; Sawyer, W. G. (University of Florida) page 43
4. **Design of Detachable Precision Fixtures which Utilize Hard and Lubricant Coatings to Mitigate Wear and Reduce Friction Hysteresis**
Culpepper, M. L.; Slocum, A. H.; Dibiaso, C. M. (Massachusetts Institute of Technology) page 47

Session IV

Interferometry

Wednesday, October 29, 2003, 10:30 AM - 12:00 PM

Session Chair: David D. Gill (Sandia National Laboratories)

1. **Fabry-Perot Displacement Metrology with a Femtosecond Laser (Invited Paper)***
Lawall, J. R. (National Institute of Standards and Technology)
2. **A Laser Speckle Sensor for Compound Rotary-linear Motion Metrology**
Shilpiekandula, V.; Trumper, D. L. (Massachusetts Institute of Technology) page 51
3. **Developing Atomic-resolution Measurements Using a Tunable Diode Laser-based Interferometer**
Zhou, H.; Damazo, B. N.; Silver, R. M. (National Institute of Standards and Technology); and Gonda, S. (National Institute of Advanced Industrial Science and Technology) page 55
4. **New Heterodyne Interferometric Method of Displacement Measure and Control with a Sub-nanometric Precision Over a Range of Several Millimeters**
Haddad, D.; Topçu, S.; Chassagne, L.; Alayli, Y. (Universite de Versailles); and Juncar, P. (BNM-INM/CNAM) page 59

Session V

Precision Machining

Wednesday, October 29, 2003, 3:30 PM - 5:00 PM

Session Chair: Thomas A. Dow (North Carolina State University), and Hashimoto Hiroshi (Kanagawa Institute of Technology)

1. **Dynamic Material Properties for Machining Simulation Using the NIST Pulse-heated Kolsky Bar**
Rhorer, R. L. (National Institute of Standards and Technology) page 63
2. **Direct Single Point Diamond Cutting of Stavax Assisted with Ultrasonic Vibration to Produce Optical Quality Surface Finish**
Liu, X. D.; Ding, X.; Lee, L. C.; Fang, F. Z.; Lim, G. C. (Singapore Institute of Manufacturing Technology) page 67
3. **Characterizing Residual Stress in Scribes on Silicon Using Deflection Measurements**
Randall, T.; Scattergood, R. O. (North Carolina State University) page 71
4. **Effect of Machining Parameters on Machining Performance of Micro EDM and Surface Integrity**
Yu, Z.; Rajurkar, K. P.; Narasimhan, J. (University of Nebraska-Lincoln) page 75

* Abstract unavailable at press time

Session VI

Advanced Machines

Thursday, October 30, 2003, 8:30 AM - 10:00 AM

Session Chair: Allen Y. Yi (Ohio State University), and Werner R. Preuss (Bremen University)

1. **Ultra-precision Surface Machining by Ion Beams and Plasma Jets (Invited Paper)***
Schindler, A. (Leibniz-Institut für Oberflächenmodifizierung e.V. - IOM)
2. **An Ultra-high Speed Spindle for Micro-milling**
Ziegert, J. C.; Pathak, J. P. (University of Florida); and Jokiel Jr., B. (Sandia National Laboratories) page 79
3. **Fundamental Analysis on the Novel 3-D Probing Technique for Microparts Using the Optical Fiber Trapping**
Hashimoto, T.; Takaya, Y.; Miyoshi, T.; Nakajima, R. (Osaka University) page 83
4. **Rotary Type Micro-Engineered Tool**
Akimoto, K.; Hashimoto, H. (Kanagawa Institute of Technology) page 87

Session VII

Novel Applications

Thursday, October 30, 2003, 10:30 AM - 12:00 PM

Session Chair: Andrew J. Hazelton (Nikon Research Corporation of America), and Aoyama Hiusayuki (University of Electro-Communications)

1. **LIGA Fabrication Technology for Micro Systems (Invited Paper)***
Henderson, C. C. (Sandia National Laboratories/Livermore)
2. **Fabrication of a Precision Mandrel for Replicating Wölter x-ray Optics**
Nederbragt, W. W. (Lawrence Livermore National Labs) page 91
3. **Pinpoint Chemical Vapor Deposition of Carbon Nanowire for Nanometer-scale Electronic Devices**
Ooi, T.; Kasuya, K.; Tsuchiya, K.; Hamaguchi, T.; Nakao, M. (The University of Tokyo) page 95
4. **Manufacturing and Quality Control of the NIST Reference Material 8240 Standard Bullet**
Whitenton, E. P.; Johnson, C. E.; Kelly, D. R.; Clary, R.; Dutterer, B. S.; Ma, L.; Song, J.-F.;
Vorburger, T. V. (National Institute of Standards and Technology) page 99

Session VIII

Equipment and Controls

Thursday, October 30, 2003, 1:30 PM - 3:00 PM

Session Chair: Stephen J. Ludwick (Aerotech, Inc.)

1. **Electromagnetically Driven Fast Tool Servo**
Lu, X.; Trumper, D. L. (Massachusetts Institute of Technology) page 103
2. **Error Compensation Using Inverse Actuator Dynamics**
Panusittikorn, W.; Garrard, K. P.; Dow, T. A. (North Carolina State University) page 107
3. **Machining of Freeform Optical Surfaces by Slow Slide Servo Method**
Tohme, Y. F.; Lowe, J. A. (Moore Nanotechnology Systems, LLC) page 111
4. **High Bandwidth Short Stroke Rotary Fast Tool Servo**
Montesanti, R. C. (Lawrence Livermore National Labs); and Trumper, D. L. (Massachusetts Institute of Technology) page 115

* Abstract unavailable at press time

Poster Session

Tuesday, October 28, 2003, 3:30 PM - 5:00 PM

Wednesday, October 29, 2003, 1:30 PM - 3:00 PM

Session Chair: Bradley H. Jared (3M Precision Optics)

BIOMEDICAL

Biomedical

1. **Fabrication of a Microrobot Movable in Flexible Pipes Like the Large Intestine**
Ono, M.; Sasazaki, T.; Kato, N.; Kato, S. (Nippon Institute of Technology) page 119

EQUIPMENT, MACHINES & INSTRUMENTS

Analysis & Modeling

1. **Design and Analysis of Nano-positioning Stage Driven by Piezoelectric Elements**
Choi, K.-B.; Ryu, S.-H.; Han, C.-S. (Korea Institute of Machinery & Materials) page 123
2. **Theoretical and Experimental Study on Surface Finish for Multi-axis CNC Milling**
Sahasrabudhe, A.; Liu, X.; Zhang, X.; Yamazaki, K. (University of California-Davis) page 127
3. **Development of an Internet-based Platform for High-speed Milling Process Parameter Selection**
Schmitz, T. L.; Tummond, M.; Duncan, G. S.; Zahner, C. (University of Florida); and Snyder, J. P. (TechSolve, Inc.) page 131
4. **Modeling and Investigation on a Jet Pipe Electrohydraulic Flow Control Servovalve**
Somashankar, S. H.; Singaperumal, M.; Krishna Kumar, R. (Indian Institute of Technology Madras) page 135
5. **Computational Accuracy Analysis of a Coordinate Measuring Machine Under Static Load**
Sousa, A. R.; Bento, D. A. (Federal Center for Technological Education - Santa Catarina) page 139

Controls

1. **Adaptive Feedforward Cancellation Viewed from an Oscillator Amplitude Control Perspective**
Cattell, J. H.; Byl, M. F.; Trumper, D. L. (Massachusetts Institute of Technology) page 143
2. **Development of an Electro-dynamic Planar Motor**
Compter, J. C. (Philips CFT); Peijnenburg, A. T. (Philips CFT NA) page 147
3. **Model-based Control Input Compensation for Parallel Kinematic Machine Tools**
Hadorn, M. C. (Institute of Machine Tools and Mfg. [IWF-ETH]) page 151
4. **Force Feedback Control of Tool Deflection in Miniature Ball End Milling**
Hood, D. W.; Buckner, G. D.; Dow, T. A. (North Carolina State University) page 155
5. **Numerical Control with Surface Driven Interpolation**
Kato, K. (Tokyo University of Science); and Takahashi, N.; Iriguchi, K. (Mitsubishi Electric Corporation) page 159
6. **Compensatory Control of Thermal Errors for High-speed Machine Tools**
Kim, K.-D.; Chung, S.-C. (Hanyang University) page 163
7. **Identification of Nonlinear Characteristics for Precision Servomechanisms**
Kim, M.-S.; Chung, S.-C. (Hanyang University) page 167
8. **A Fuzzy Logic Based Adaptive Feedforward PI Controller for Nanometer Positioning**
Lu, X.; Tran, H. D. (University of New Mexico) page 171
9. **Waveform Determination and Positioning Control of the NanoSlider with Inertial Sliding and Stick-positioning**
Shim, J. Y.; Gweon, D.-G. (Korea Advanced Institute of Science & Technology - KAIST) page 175
10. **Development of Controlled Surface Acoustic Wave Planar Actuators**
Vermeulen, M. M. P. A.; Peeters, F. G. P.; (Philips CFT); and Peijnenburg, A. T. (Philips CFT NA) page 178

Design & Testing

1. **Differentiating Guidance from Support Functions in Ultra Precision Mechanism Design****
Gee, A. E. (University College, London) page 182
2. **Ultraprecision XY Stage with Millimetre Travel Range Using Leaf-spring Mechanism and Voice-coil Motor**
Kang, D.-W.; Kim, D.-M.; Kim, K.-H.; Shim, J.-Y.; Gweon, D.-G. (Korea Advanced Institute of Science and Technology - KAIST) page 183

**Extended abstract unavailable at press time

3. **Control and Design of a Dual Servo with Fine 6-axis Stage**
Kim, K.-H.; Lee, M.-G.; Kim, D.-M.; Gweon, D.-G. (Korea Advanced Institute of Science and Technology) page 187
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Mishima, N. (National Institute of Advanced Industrial Science and Technology) page 195
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Nakao, Y.; Mimura, M.; Kobayashi, F. (Kanagawa University) page 199
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Werkmeister, J. B.; Slocum, A. H. (Massachusetts Institute of Technology) page 207
9. **Experimental Determination of Kinematic Coupling Repeatability in Industrial and Laboratory Conditions**
Willoughby, P. J.; Hart, A. J.; Slocum, A. H. (Massachusetts Institute of Technology) page 211

Mechatronics

1. **Design and Development of a High Precision Lens Focussing Mechanism Using Flexure Bearings**
Gaunekar, A. S.; Widdowson, G. P.; Srikanth, N.; Guangneng, W. (ASM Technology Singapore Pte. Ltd.) page 215
2. **Synthesis of the 3D Linux-based CNC System for Precision Machines**
Kang, B.-K.; Kim, M.-S.; Chung, S.-C. (Hanyang University) page 219
3. **Fabrication of an Inch-worm Type Mobile Microrobot Movable in Different Diameter and Long Pipes**
Kato, S.; Ono, M.; Aizawa, Y.; Nakamura, H. (Nippon Institute of Technology) page 223
4. **Fine Motion Performance of L-shaped Seal Mechanism with 3 Degrees of Freedom**
Kawagoe, K.; Furutani, K. (Toyota Technological Institute) page 227
5. **Design and Development of Micro Hopping Robots for Rescue Operation**
Misumi, R.; Kurata, S.; Aoyama, H. (University of Electro-Communications) page 231
6. **A Simple Hybrid Positioning Stage With <1nm Resolution**
Okazaki, Y. (National Institute of Advanced Industrial Science and Technology);
Ichikawa, S. (Sigma Tech Co. Ltd.); and Otsuka, J. (Shizuoka Institute of Science and Technology) page 235
7. **Fabrication of an Artificial Earthworm Type Mobile Inspection Robot Movable in 100 m Long Pipes**
Ono, M.; Kato, M.; Naito, T.; Sasazaki, T.; Kato, S. (Nippon Institute of Technology) page 239
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Technical Papers



A S P E

Achieving Precision through System Engineering

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Introduction

Creating mechatronic equipment with performance that advances the state of the art in the industrial world is a field where Philips CFT has been engaged in for over 30 years. Building on expertise from development of manufacturing equipment, high speed and high accuracy, we tracked the different generations of automation and mechatronics. During these years a system-oriented approach was established based upon the design of successive generations. In this paper we will describe the development of our approach and give our view on the next stages of development.

System approach

In mechatronic systems we are generally dealing with the challenge to control the motion or position of mechanical parts. To achieve this control goal, measurements are taken and compared with the required values generated by a controller. The observed deviations are used to drive actuators to correct the differences between required and actual motion or position.

The simple mechanical system shown in figure 1 is designed to control the position of the child in the water such that the mouth will be above the water surface. In the design of such systems four distinct steps can be distinguished ;

1. Determination of the requirements.
2. Estimation of the disturbing effects.
3. Calculation of the desired positioning power
4. Design of the different system elements.

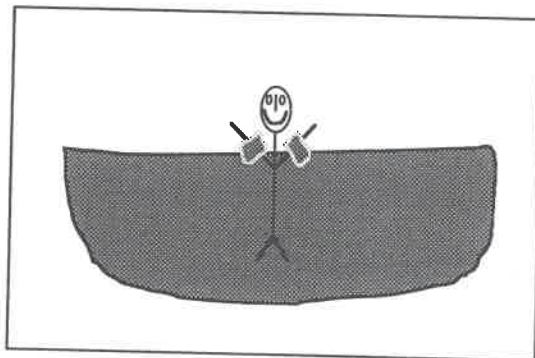


Figure 1. A simple representation of a positioning system where the positioning device has been designed to maintain the required position in the presence of different disturbances.

In the first generations we were confronted with systems lacking power to control a required position. The disturbing forces or motion of the targets were too large so that the required position could not always be achieved.

Therefore the first phase of our systems oriented approach was directed at achieving larger controlling power.

In the later phases we directed our attention to a more extensive analysis of the disturbances and to a more integral handling of the requirements.

The Power Design Approach

The power of a mechatronic system to maintain its required position is very much similar to the mechanical stiffness in passive systems. A disturbing force should not lead to a large position error. Therefore a design with a high stiffness is required. When dynamics are involved, the acceleration of masses will lead to disturbing forces. When these forces are small enough the position errors will also remain small. Combining the high stiffness and low mass leads to higher values for the natural frequencies of vibration.

For controlled systems the arguments are similar and we strive for high values of the bandwidth of the feedback controller. High loop gain through low masses and powerful feedback may lead to high bandwidths. Increasing this bandwidth however is limited due to contributions from all elements in the feedback chain.

From a mechanical design perspective, the presence of undesired vibrations provides a severe limitation. Our work on optical disc drives and wafer steppers has forced us to develop a profound understanding of these issues and to find effective methods to handle the vibrations.

In figure 2 the deformed shape of an actuator for an optical disc drive is shown. This actuator was in production for Audio CD application requiring a bandwidth of about 1000 Hz. The introduction of CD-ROM applications made it necessary to increase this bandwidth to about 3000 Hz. Due to a mechanical vibration at about 10 kHz this was not directly achievable. Conceptual changes were not acceptable because of short lead times for the product and the costs associated with that.

Based upon a thorough understanding of the dynamics involved, a solution was obtained where we reduced the mechanical stiffness and increased the moving mass of the actuator. With minor adjustments in the product we managed to build a successful generation of CD-ROM modules.

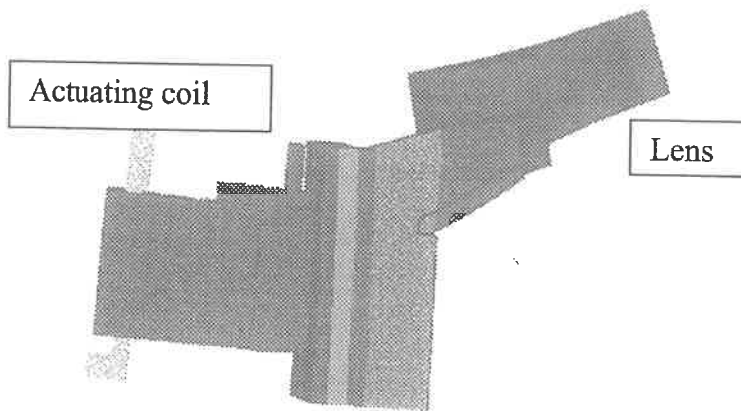


Figure 2. Deformed shape of the moving part of an optical disc lens actuator. Actual size is about 15 x 8 X 8 mm. while the moving mass is about 0.7 gr.

Eliminating Disturbances

Optimizing the design of the different elements in the feedback loop is limited and at a certain point in time it proved to be more economical to fight the disturbing sources and transmission paths. In figure 3 a simple lumped parameter presentation for a wafer stepper with a linear motor drive is shown. Floor vibrations form one of the disturbing effects. Such vibrations are transmitted through the machine base and lead to motion of the optical system. The controller must create forces to make the wafer follow these motions. Initially, the filtering frequency of the air mount suspension for the machine was about 3 Hz. Introduction of active vibration control technology in 1992 allowed for reduction of that frequency to about 1 Hz, yielding an improvement with a factor 10 for allowable levels of floor vibrations.

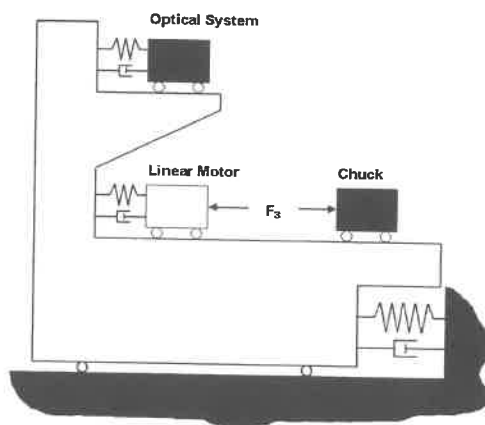


Figure 3. One DOF lumped parameter model for a wafer stepper. The chuck carries the wafer and must be positioned relative to the optical system. The linear motor drives the chuck.

In vibration isolation systems the reduction of the suspension frequency is beneficial for improving the isolation. Unfortunately low stiffness of the suspension makes it very sensitive to disturbing forces acting on the payload. To circumvent this effect we have developed a novel isolator system that combines high stability against disturbing forces with the extreme isolation performance of low stiffness suspensions.

One example of forces that are disturbing the payload is the reaction force from the linear motor. When high accelerations are needed for competitive performance, large driving forces are required. Such forces will excite the metrology structure and deteriorate the positioning accuracy. For the ASML lithography tools we introduced actuation concepts that avoid such excitation allowing for the combination of high speed and high accuracy. A similar solution is shown in figure 4 where two linear motor driven gantry stages work in one machine. When one stage accelerates at full power the other stage must maintain its positioning accuracy. A clever integration of two frames has allowed us to achieve this without requiring additional "power" of the feedback control.

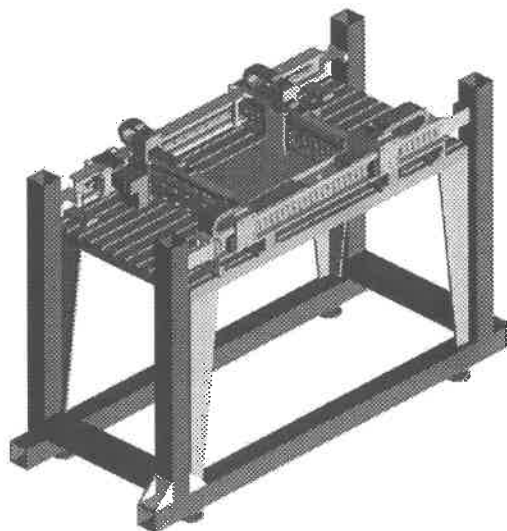


Figure 4. A functional model for a novel machine concept for SMD-placement where two frames are intertwined to separate driving forces from metrology components.

Managing the System Requirements

At the start of the development the requirements are determined. Initially the development teams take the specifications for granted and make designs based on the previous steps we described. Once the systems reach the limits in “power” and all disturbing effects have been optimally cancelled, insufficient performance (compared to the specifications) urges to look for new ways to satisfy the demands as set by the customer, or product management. It turns out that it is quite profitable to spend effort in setting up system error budgets. And make sure to include all parts of the system in this analysis. In many cases the customer, or product management, has done a top-level distribution of precision requirements. It pays of handsomely in many cases to reassess this distribution.

One example of such a system approach can be seen in the first generation wafersteppers from ASML. In these tools overlay accuracy of about 50 nm was achieved over a 6” wafer. Basically this requires a sensing system with accuracy of 0.3 part per million. This requirement is not easily or economically met in an industrial environment. Through a well-designed process of initializing the sensing system and imaging process a significant reduction of the requirements for the sensing system has been achieved.

Similar performance aspects can be found in the concept of the Ultra-Precision CMM developed by Theo Ruijl from Philips CFT. Rigorous application of Abbe’s rules, a description of a calibration strategy at the start of system development and extreme synchronization of data acquisition form the basis for high performance at an industrial cost level. This work forms the basis for the product developed by IBS and Philips CFT.

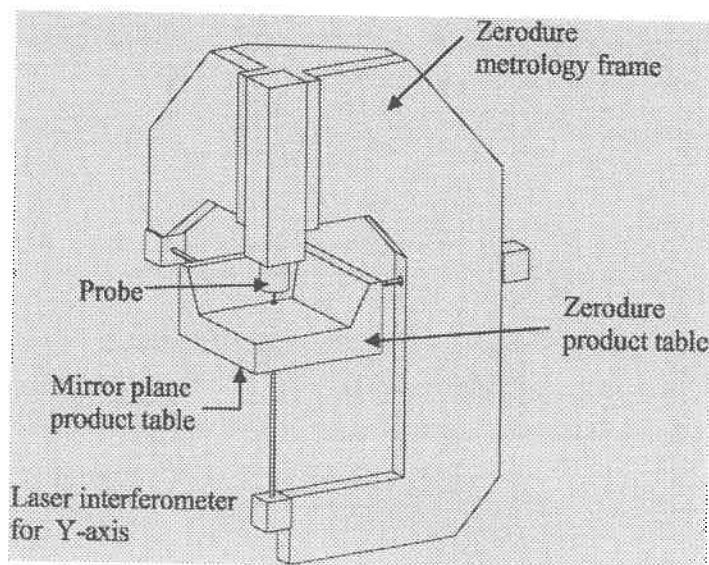


Figure 5. Schematic diagram of the metrology concept for the Ultra-Precision Coordinate Measuring Machine.

Future Developments

The different development steps in our approach to System Engineering for High Precision systems will be followed by new advancements. In our environment we are now facing the challenge to improve the reliability of project planning for highly innovative projects. Continuous learning in this respect has given us a good basis for this.

From a technology viewpoint we are closely following developments of adaptive and learning machines. These fields appear to be highly promising to shift the limits of performance ever further.

Application of our experience in the design of Micro Systems forms an additional field of activity. However the scale, manufacturing, test and packaging pose new challenges in addition to the current ones. Given the wealth of process improvements we think that the experience of good system design will be the basis for building the industrial success of these devices.

MICROTECHNOLOGY-BASED ENERGY AND CHEMICAL SYSTEMS AND MULTI-SCALE FABRICATION

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ABSTRACT

Microtechnology-based Energy and Chemical Systems (MECS) are devices which process bulk amounts of fluids within networked configurations of thermal, mechanical and chemical unit operations each based on highly-parallel arrays of microchannels. These miniaturized energy and chemical systems enable large quantities of fluid to be processed in small amounts of space and, in Europe, the application of MECS technology is called process intensification. The use of microchannels results in shorter diffusional distances which permit shorter residence times in channels, shorter channel lengths and, consequently, more compact systems. All of this is accomplished with no pressure drop penalty. As a result, MECS devices have the potential to revolutionize the processing of mass and energy where a premium is placed on mobility, compactness, or point-of-use production. This paper reviews some of the underlying concepts that enable MECS technology, considers the impact of this technology on society, the means by which they are made and, as a consequence, considers the role of the manufacturing engineer.

Keywords: microtechnology, energy systems, chemical systems, process intensification, manufacturing engineer.

INTRODUCTION

Over the past 40 years, there has been a growing emphasis in manufacturing research on the fabrication of devices with multiple length scales. Multiple length scale systems, or simply multi-scale systems, are defined as systems with feature sizes in at least two different length scale regimes (i.e. macro, meso, micro, nano, etc.) for the improved physical or chemical system performance. Examples of multi-scale systems include integrated circuits, microelectromechanical systems (MEMS), and micro total analysis systems (μ TAS). For example, within microelectronics, nano-scale gates are embedded within micro-scale circuits that are connected to meso-scale bonding pads and ultimately macro-scale chip-scale packages (CSPs) and printed circuit boards (PCBs). Smaller gate sizes permit higher component densities improving both speed and cost. At the same time, meso-scale pads permit electrical interconnection while CSPs and PCBs simplify handling and foster component modularity.

One emerging class of multi-scale systems is Microtechnology-based Energy and Chemical Systems (MECS). MECS are highly-parallel microfluidic devices that rely on embedded nano and micro-scale features to improve heat and mass transfer. As shown in Table 1, the key distinguishing feature of MECS is that they use microchannels to process *bulk fluids*. By contrast IC's, MEMS and μ TAS devices are used in some fashion to generate or process *information* whether in signal acquisition, signal processing or knowledge acquisition. Most microfluidic applications (e.g. μ TAS) involve processing nanoliters or even picoliters of fluids for the purpose of conducting lab-on-a-chip type experiments and, ultimately, to extract *knowledge* or *information* from these experiments. However, MECS devices are used to process bulk amounts of *fluids* such as tanks full of gasoline (fuel reformers) or rooms full of air (distributed HVAC). Consequently, as processors of *mass* and *energy*, MECS devices are relatively large.

To process large quantities of fluids within microchannels, the microchannels must be arrayed. Therefore, most MECS devices typically involve the integration of one or more microchannel arrays, called unit operations (e.g. micromixer, microseparator, microreactor, microheat exchanger, etc.), into a microsystem. Examples of MECS unit operations include microchannel heat exchangers, which have demonstrated heat fluxes 3 to 5 times higher than that of conventional heat exchangers.^{1,2} Other examples include microreactors for waste remediation consisting of nanocrystalline catalysts embedded within the walls of a microchannel array connected to macro-scale fluidic connectors. Examples of application areas for MECS microsystems include advanced climate control³, fuel processing⁴, and microdialysis⁵ among others.

UNDERLYING CONCEPTS

MECS devices are primarily focused on taking advantage of the extremely high rates of heat and mass transfer available in microchannels to radically reduce the size of a wide range of energy, chemical and biological systems. Typically a MECS device will have dimensions on the order of 1 to 50 cm but it will include embedded, highly-parallel arrays of microchannels where individual channel dimensions are on the order of 100 μ m in height by

Table 1. A comparison of four different types of multi-scale systems.

Parameter	Integrated Circuits (ICs)	MEMS Devices	Micro Total Analysis Systems (μ TAS)	MECS Devices
Primary function of device	Information/signal processing	Mechanical actuation /information/signal processing	Analytical processing i.e. 'Lab on chip', 'Fluid in-Answer out'	Bulk fluid processing device-heat/mass transfer
Materials used	Semi-conductors & ceramics. Metals used for interconnects	Metals, Semi-conductors, polymers and ceramics	Metals, Semi-conductors, polymers and ceramics- Biocompatibility is a key issue	Mostly limited to metals and polymers due to high operating temperatures
Primary components of system	Transistors, switches	Mechanical device for motion, electronic device for signal processing	Fluid processing, electrical, mechanical and optical systems	Micro-channels for heat and mass transfer, surface reaction
Device Scale (Size of full device)	Dies range from few mm to few cm-square or rectangular dies	Sizes range from few mm to few cm	Few centimeters to few inches	Few mm to several hundred centimeters
Feature Scale (E.g.: transistor size in IC)	Highly microscopic: Features range from tens of nm to 0.25 microns	Microscopic: Few microns to several hundred microns	Multi-scale: Few microns to few mm	Multi-scale: μ channels are 20 to 250 microns; catalysts, nanocoatings and pores are submicron.
Device Tolerances	Alignment tolerances are in the nm ranges for high-end IC's	Several hundred nm range	Few microns to several hundred microns depending on size	Few microns to several hundred microns depending on size
Fabrication Methods	Lithography techniques (additive/subtractive), Silicon micromachining	Lithography, surface and bulk micromachining	Multiple modes of fabrication exist due to variety of components on system	Microlamination and limited micro-investment casting
Applications	Processors, memory cards, video cards, etc (general electronics)	Bubble jet print heads, micro-mirrors, airbag sensors, accelerometers, etc	Bio-MEMS, DNA and tissue analysis, gas chromatographs, etc	Fuel processors, microheaters, portable power generators, etc
Benefits	Small devices, Fast Processing-Can be integrated into other components	Fast response of mechanical devices coupled with information processing.	Enhanced portability, decreased analysis time, small sample volume	High rates of heat and mass transfer, portable, decentralized fluid processing

several mm in width. (A human hair has a diameter of approximately 80 μ m.) By taking advantage of the enhanced rates of heat and mass transfer in the embedded microchannels, researchers have typically been able to reduce the size of a variety of energy and chemical systems by a factor of 5 to 10.^{1,2,6,7}

How does this all work? Single-phase flow in a 100 μ m high channel will, in almost all cases of interest, be laminar, with Reynolds numbers typically between 1 and 100. In laminar flow, the residence time required to have the fluid reach thermal equilibrium with the walls of the channel decreases as D^2 where D is proportional to the height of the channel. If we decrease the channel height by a factor of 10, we will decrease the required residence time by a factor of 100. This means that, in theory, a heat exchanger with 100 μ m high channels would be 1/100th the size of a heat exchanger with 1 mm high channels for the same heat transfer rate and it can be shown that for laminar flow, the increased pressure drop associated with the smaller channels is exactly off set by the decreased length of the channel required to deliver the same heat transfer rates. In effect we get a theoretical factor of 100

reduction in size with no increase in pressure drop. Ultimately, by going to smaller channels we will reduce the diffusion barrier to the point where other processes such as axial conduction along the heat exchanger determine residence time for thermal equilibrium. At that point, further reductions in channel dimensions will not result in reduced residence time.

While single-phase flow in microchannels is well understood, processes that involve wall effects such as phase change and surface tension are significantly different in micro-scale geometries. Boiling is an example. The limited experimental investigations of boiling in microchannels suggest that the process of phase change in a microchannel is different from macro-scale devices. However, published performance for microchannel evaporators and condensers show a significant improvement in performance when compared to single phase microchannel devices, suggesting that as our understanding of phase change in microchannels improves we can expect a further improvement in the performance of microchannel heat exchangers.

Mass transfer is similar to heat transfer in microchannels. The residence time that reactants need to reach complete conversion in a catalytic gas phase reactor also decrease by D^2 if the residence time is dominated by diffusion rather than reaction kinetics. Many important reactions such as steam reforming are catalytic gas phase reactions where the primary barrier to short residence time is the time it takes for the reactant to diffuse to the catalyst site and for the products to diffuse back into the bulk flow. Reductions in residence time and reactor size on the order of 20 times have been reported in the literature.⁶ Ultimately, by going to smaller channels we will reduce the diffusion barrier to the point where reaction kinetics will determine residence time for complete chemical conversion. At that point, further reductions in channel dimensions will not result in reduced residence time.

TECHNOLOGICAL IMPACT

The advancements described above will allow the development of more compact energy and chemical microsystems that could significantly impact the way in which mass and energy are processed and consumed in society. Examples of current MECS microsystem developments at Oregon State University include man-portable heat pumps, on-board automotive fuel reformers for fuel cells, and dime-sized, wearable biosensors for detecting toxins. In the future, chemical microreactors for producing bleach or laundry detergent from more concentrated, benign feedstocks may be a standard feature on every washing machine usurping vertical supply chains for the production of household chemicals. These new technologies will have dramatic effects on the distribution of goods in industry as the processing of mass and energy becomes more decentralized.

Successful development of MECS technology will enable the development of a new class of miniaturized energy, chemical and biological systems. Examples of systems and products that can result from MECS include: compact cooling systems for electronics and aviation applications; distributed hydrogen production for automotive fuel cells; high energy density batteries for manportable power and micro-scale power applications; distributed, residential space conditioning; exhaust-driven automotive cooling; man-portable cooling; catalytic microreactors for waste cleanup; in-situ resource processing for space exploration; and CO₂ remediation and sequestration among others.

MULTI-SCALE FABRICATION

Much of the promise of MECS depends on the availability of economical methods of production that allow MECS to compete with the cost of existing centralized energy and chemical systems (e.g. HVAC). While MECS devices may include features with nano-scale or micro-scale dimensions, the development of the technology has emphasized materials and fabrication techniques that are significantly different from conventional silicon-based micromachining. MECS devices are implemented mostly in metals, ceramics and selected polymers due to a need for thermal and chemical properties in applications ranging from man-portable heat pumps and automotive fuel reforming. In contrast, most of information processing miniaturization technologies such as IC, MEMS and μ TAS devices are based on semiconductors.

In particular, microlamination architectures have been offered as one approach to the economical production of MECS devices.⁸ As shown in Figure 1, microlamination involves the patterning, registration and bonding of thin layers of material, called laminae, to produce monolithic microsystem devices with embedded, high-aspect-ratio microchannel arrays. The production of energy and chemical systems with microlamination schemes reduces the issue of energy and chemical system implementation from centralized plant construction to discrete part manufacturing which is the domain of the manufacturing engineer. Therefore, one outcome of the trend toward

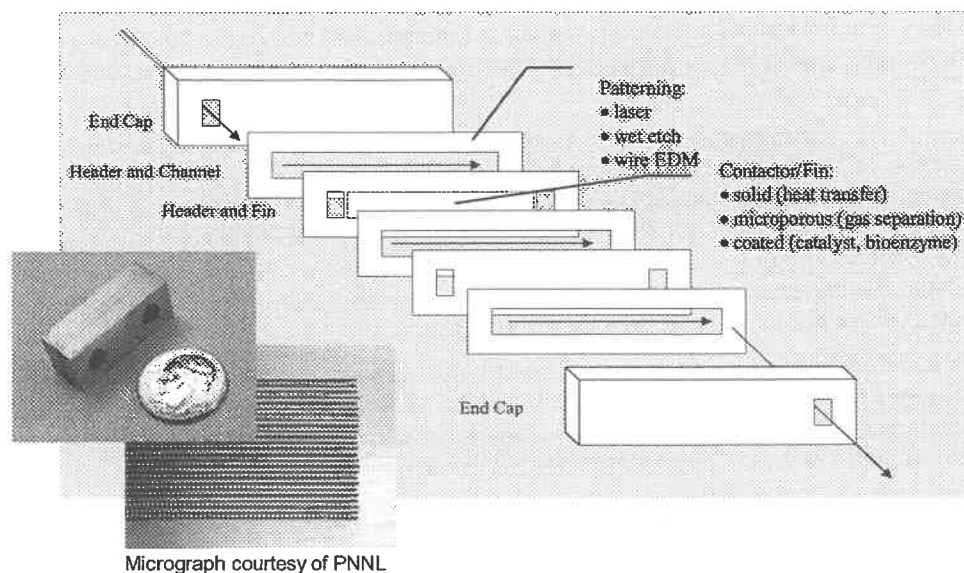


Figure 1. Conceptual presentation of a microlamination architecture.

energy and chemical systems miniaturization will be a growing dependence of the chemical and energy industries on the manufacturing engineering discipline.

Although microlamination architectures are currently the basis for producing commercial products, very little is known about the cost and technology drivers involved. To date, most MECS work has been performed in the prototyping and laboratory environment. As a result, a focus of future manufacturing research efforts will be to study the economics of microlamination. Also, it is proposed that more research is needed to understand the mechanisms of warpage and dimensional stability within microlaminated devices. These and other issues related with MECS production are the domain of the manufacturing engineer.

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High-NA interference microscopy of complex surface structures

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INTRODUCTION

Perhaps the most versatile interference microscope for <1-nm repeatability surface profiling relies on scanning white light interferometry (SWLI), for which a spectrally broadband or “white light” illumination localizes fringe contrast to near zero path difference (Figure 1). The data acquisition scan ζ in a direction orthogonal to the part surface accommodates step heights and other discontinuous features while handling rough surface textures that generate complex speckle patterns without fringe order uncertainty.

Recent advances in SWLI include dual-sided measurements for flatness, thickness and parallelism of microscopic parts, flat panel displays, stroboscopic interferometry of objects in motion, and the measurement of thin film structures. In many of these applications, the high numerical aperture (NA) of microscope objectives plays an important role. The following examples illustrate the interesting and sometimes surprising nature of high-magnification interferometry of complex surface structures.

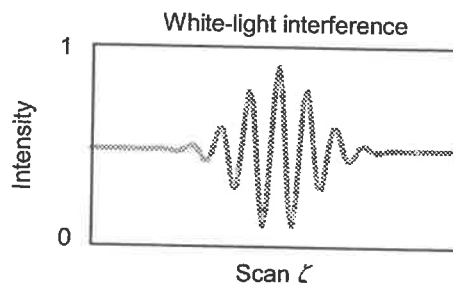


Figure 1: Interference pattern for a short-coherence interferometer.

HIGH NA RELATIONAL METROLOGY OF MICROSCOPIC PARTS

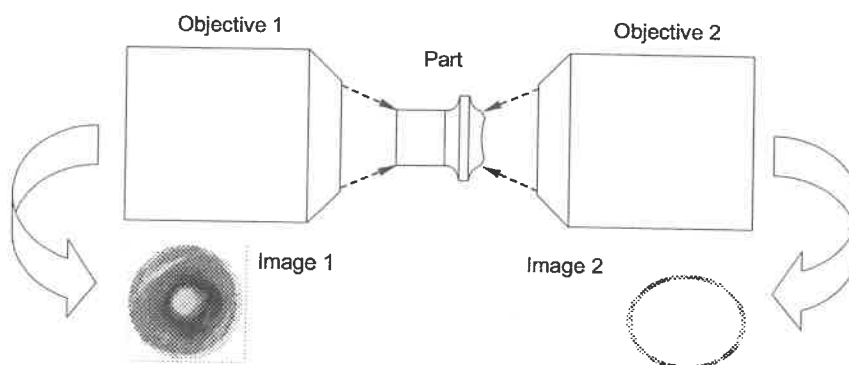


Figure 2: Simplified schematic of an instrument that simultaneously measures the form, separation and relative orientation of two sides of a part having complex surface features, including a rounded edge on the right side that requires high NA imaging.

In optical metrology, we are accustomed to measurements of surface form using interferometry. Less familiar are relational measurements of part dimensions such as thickness and parallelism—important parameters in industrial applications where systems comprise assemblies of tightly fitting parts. Such measurements require multiple fields of view,¹ while some of the more challenging applications involve complex surface features that are difficult to accommodate in a general way at low NA. Here the high NA of a microscope objective, combined with the SWLI principle, make it possible to capture light and generate a surface profile from light that is reflected or even scattered at high angles.² We have constructed an instrument for this purpose using the basic geometry shown in Figure 2. The automated version of this tool shown in Figure 3 measures the distance from a flat surface to a best-fit circle to a curved surface, as well as other parameters, at the rate of 5 seconds per measurement.

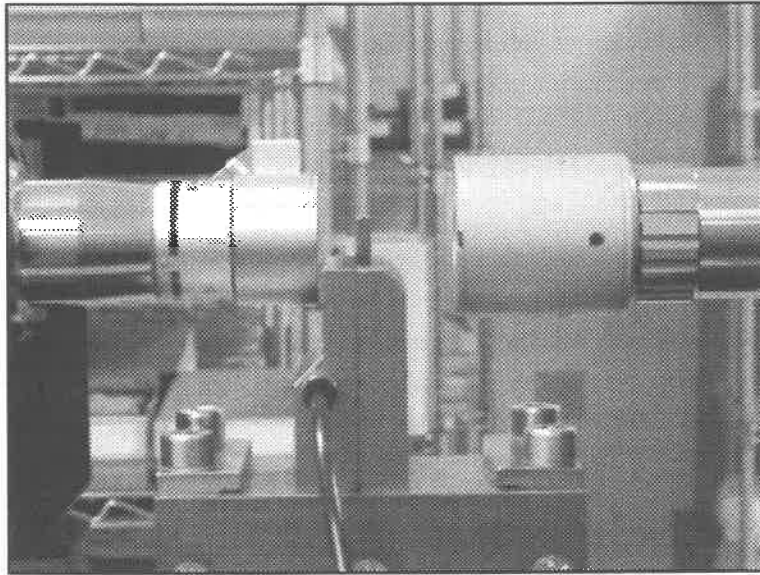


Figure 3: Photograph of a dual-sided SWLI microscope for production-line verification of microscopic parts.

HIGH NA MONOCHROMATIC MEASUREMENTS

The SWLI principle is most often associated with the spectral properties of the source light. However, when the NA of the objective is large, there is an equally important spatial coherence effect that also localizes fringes. For even a moderate NA of 0.6, one can perform the equivalent of SWLI with narrow-band or even monochromatic light (Figure 4). Potential benefits include a high-resolution UV-laser based microscope, and hybrid systems that determine optical properties of materials by comparing high NA and broadband measurements of the same sample.³

Another benefit of high NA of interference microscopy is the ability to use long-life LED or laser light sources that can be modulated rapidly for stroboscopic illumination of dynamically actuated objects (Figure 5).^{4,5,6} The NA introduces a range of incident angles on the sample surface, effectively introducing a range of equivalent wavelengths proportional to the cosine of the incident angle. Note however that these objective obliquity effects and the rapid LED modulation introduce wavelength uncertainty, which can be overcome by using a frequency-domain analysis technique that uses the scan rate as the height metric, rather than the wavelength.⁷

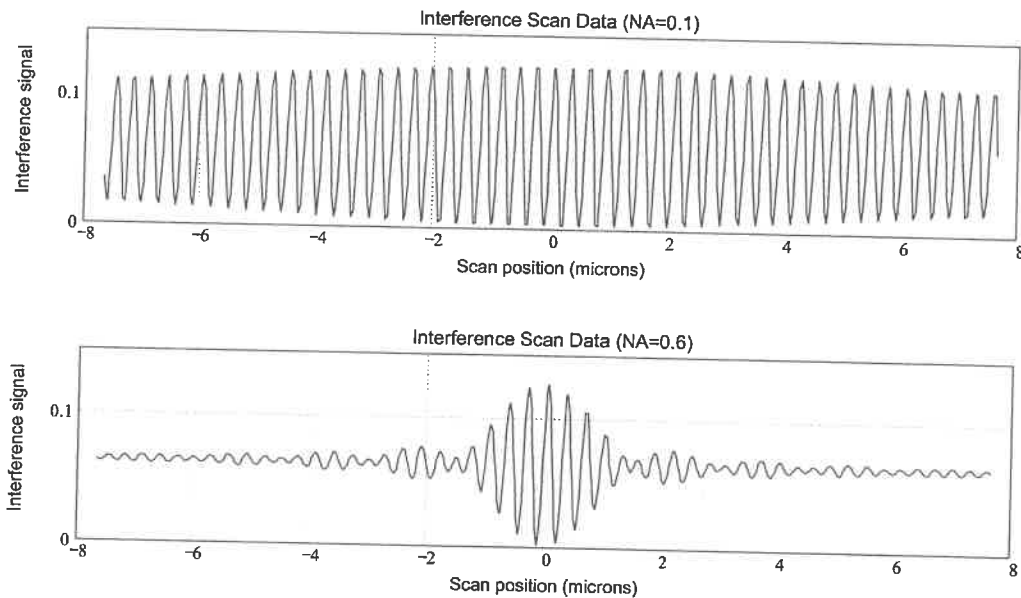


Figure 4: Simulation illustrating fringe contrast localization at high NA for a narrow-band light source ($\lambda=600\pm 10\text{nm}$). Upper figure: NA=0.1; Lower figure: NA=0.6.

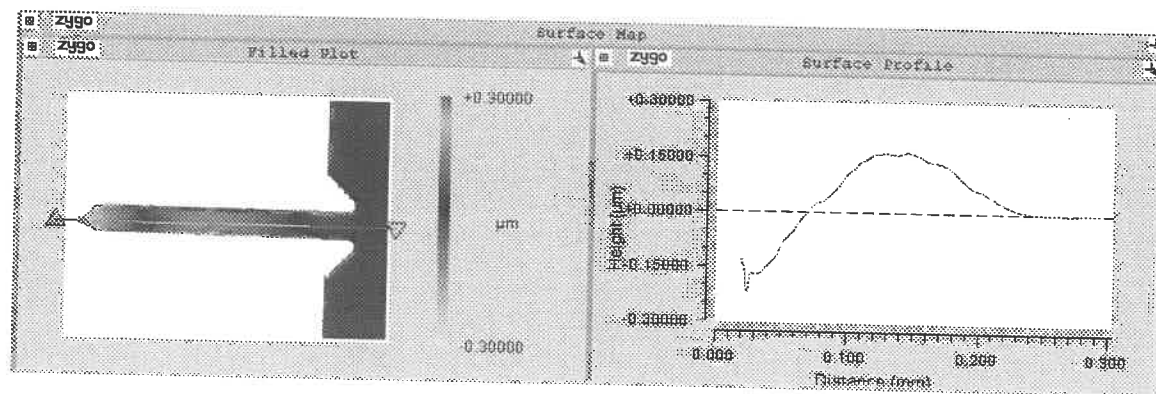


Figure 5: SWLI profilometry of a micro cantilever oscillating at 75kHz, captured by a strobed LED light source.

THIN FILMS

Important sample types, such as flat panel displays, are built up from thin transparent film layers. When profiling this type of structure, one often sees a sequence of signals during a ζ scan corresponding to the interfaces between transparent media. This phenomenon can be used to measure the thickness of the films themselves,⁸ or as is more often the case, to distinguish between the top-surface signal of interest and the unwanted echoes from underlying structure.

When measuring thin film structures, one observes that the underlying layers usually appear *deeper* than they actually are, because the index of refraction increases the optical path delay. However, a high NA objective tends to localize the contrast of internal reflection at a *shallower* level, because of refraction. When both high NA and broad bandwidth are

present, the two effects compete, leading to a beneficial suppression of the signals coming from the underlying layers. Thus by combining high NA with broad bandwidth illumination, automated software easily identifies the top-surface signal and rejects the now much weaker echo (Figure 6).

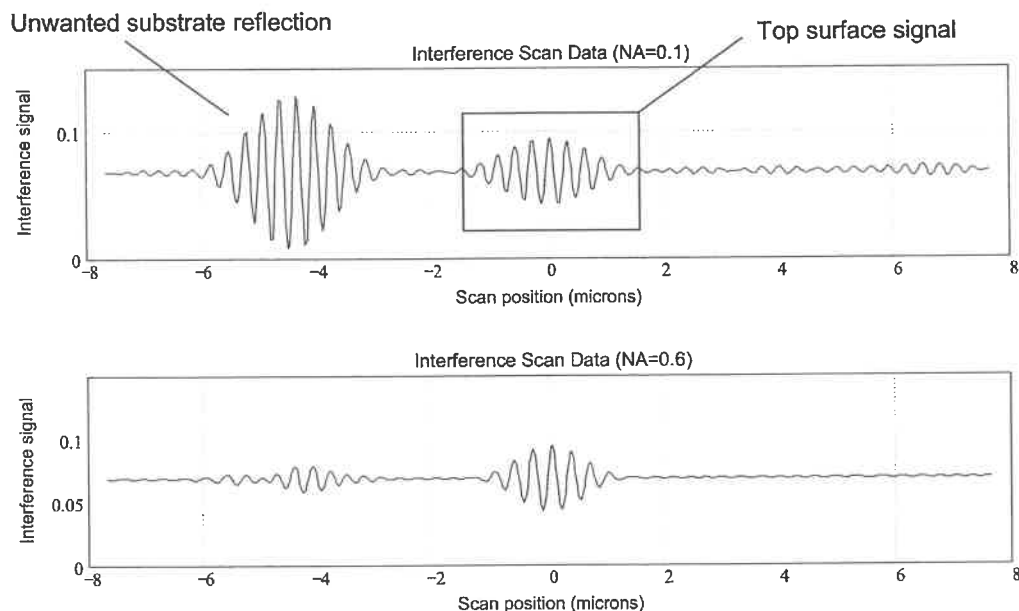


Figure 6: Simulation illustrating suppression of an unwanted signal from an underlying thin film interface using a combination of high NA and broad spectral bandwidth. The surface model is 3 microns of SiO₂ on Si, $\lambda=600\pm100\text{nm}$.

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Interferometric thickness measurement of free form silicon wafers

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ABSTRACT

A relatively low cost pre-prototype measurement machine for measuring the geometry of double side polished wafers has been developed. The measurement principle is based on a scanning double side Fizeau interferometer with which the front side flatness and the back side flatness of a wafer is measured simultaneously. Both flatness maps are used to calculate the wafer thickness variation. The influence of lens aberrations, alignment errors and flatness errors of reference flats is compensated for by self-calibrating software compensation techniques.

Keywords: thickness measurement, scanning interferometer, self calibration

1. INTRODUCTION

For characterization of wafer thickness and flatness many challenges arise. Capacitance gauging techniques will give problems measuring wafers of strongly varying conductivity. Transmissive thickness measurements using an infrared interferometer will fail for non-transparent highly doped wafers. A solution for measuring these wafers is found in simultaneously measuring the front side and back side flatness of a wafer using a Fizeau interferometer. With both flatness maps aligned accurately a thickness map can be derived. Since the price of optical components tends to grow exponentially with size, the cost of ownership for a full size double side interferometer for measuring up to 300 mm silicon wafers is enormous.

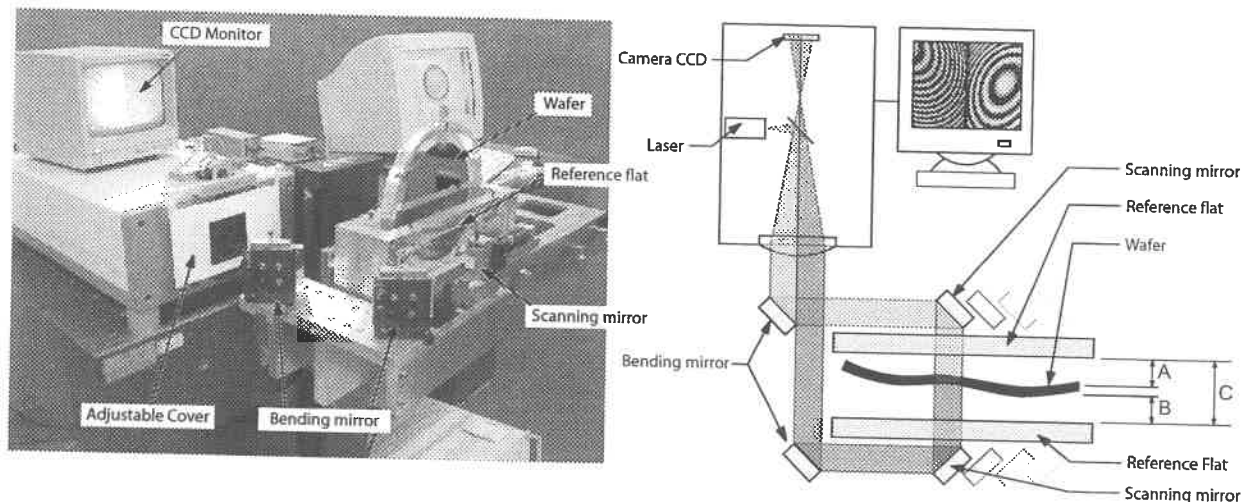


Figure 1. Left: Pre-Prototype measurement setup, Right: A schematic view(right)

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To overcome the described difficulties, the Eindhoven University of Technology has developed a scanning double side surface interferometer for measuring the flatness and thickness variation of up to 300 mm double side polished wafers. The described calibration procedure makes the thickness measurement insensitive to wafer tilt and bow.

2. MEASUREMENT PRINCIPLE

In figure 1 the schematic setup is shown. The parallel measurement beam from a stabilized HeNe laser is split into a left part and a right part. The left part measures the distance between the rear reference flat and the back side of the wafer (*A*), while the right part measures the distance between the front reference flat and the front side of the wafer (*B*). The thickness variation of the wafer can be derived from the flatness measurements (*A* and *B*) and the calibrated distance between the reference flats (*C*).

$$\text{Wafer thickness} = C - A - B \quad (1)$$

Both sides of the wafer are measured simultaneously. The height difference between neighbouring fringes equals half the wavelength of the light. The flatness maps can be determined by means of a phase shifting method or by means of a spatial carrier fringe method^{1,2}. For the phase shifting method multiple phase shifted interferograms are recorded. The phase shift is introduced by translating the reference flats relatively to the wafer by means of a piezo element. By tilting the wafer a spatial carrier fringe method can be used to obtain the flatness of the wafer on both sides from only a single interferogram. The pre-prototype version is a step and measure instrument, which measures a surface area of 25x45 mm at a time. Horizontal scanning is done by moving the scanning mirrors alongside of the rectangular reference flats. The vertical scanning is done by moving the wafer in between the reference flats. This way a full coverage of wafers up to 300 mm diameter is obtained. To reduce the fringe density while measuring heavily bowed wafers, the machine has a tilt adjusting mechanism. The wafer is locally aligned to be approximately parallel to the reference flats at all times. As a result the amount of fringes is minimized and the alignment specifications of the front side flatness map on the back side flatness map for obtaining thickness is reduced.

3. THICKNESS CALIBRATION

To determine the thickness variation of a wafer the distance variation between the two reference flats is measured (term *C* of eq. 1). For this the wafer is positioned outside the viewing area of the interferometer. Depending on which part of the beam from the interferometer is covered interferograms will appear as shown in figure 2. When the left side of the interferometer is covered the distance between the reference flats is measured from the

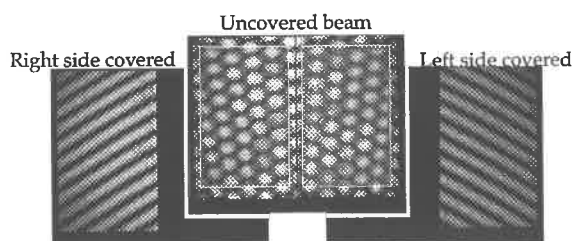


Figure 2. Reference flat measurement (left or right side covered), pre-alignment measurement (uncovered)

front side. When the right side of the interferometer is covered the same distance is measured from the back side. Therefore the left and the right images of figure 2 are each other's mirror. While making a horizontal scan multiple interferograms are recorded. A thickness reference map can be calculated from the recorded interferograms by using a spatial carrier fringe algorithm (figure 3). This scanning calibration measurement of distance *C* can be done periodically to eliminate the effect of thermal or mechanical drift. When no wafer is present and the interferometer is not covered a mixture of interferograms appears (figure 2). The symmetry of

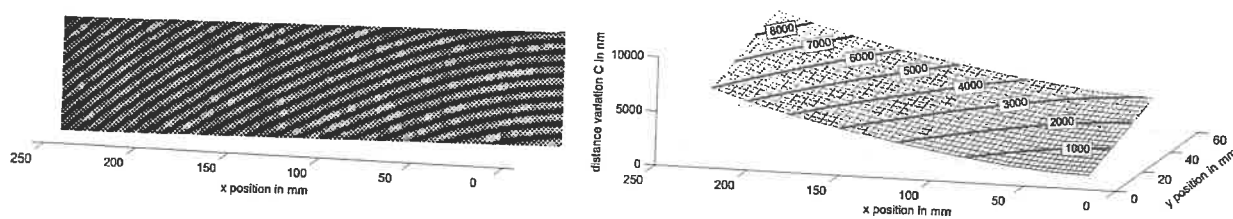


Figure 3. Left: Stitched interferograms, Right: Calculated thickness reference values

these mixed interferograms is observed to make a pre-alignment of the front side flatness map on the back side flatness map.

To determine the effect of alignment errors, lens aberrations, scaling errors and rotation errors the wafer is measured at varying horizontal and vertical tilt positions (left of figure 4). Principally the derived wafer thickness should be independent of tilt or bow of the wafer surface. Due to the mentioned error sources however the true wafer thickness is not measured correctly. Figure 4 shows the relation between the measured

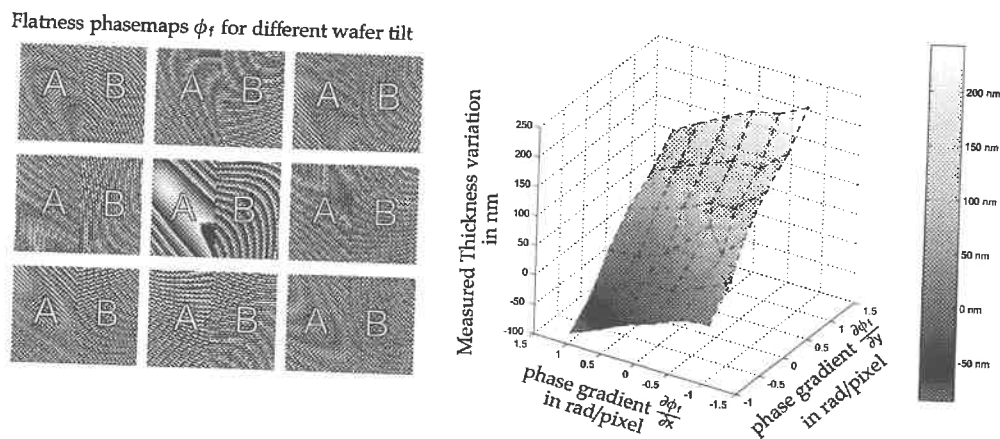


Figure 4. Left: Recorded phasemaps at different wafer tilt positions, Right: Measured wafer thickness for varying gradients of the front side flatness phasemap ϕ_f at a specific CCD pixel coordinate.

thickness and the phase gradient of the local flatness map ϕ_f . It is assumed that the true thickness is measured when there is no tilt of the wafer ($\frac{\partial \phi_f}{\partial x} = 0$ and $\frac{\partial \phi_f}{\partial y} = 0$). Then the relationship is described well by the linear equation 2:

$$\text{measured thickness} = \text{true thickness} + a \frac{\partial \phi_f}{\partial x} + b \frac{\partial \phi_f}{\partial y} \quad (2)$$

The calibration parameters a and b of equation 2 can be estimated by fitting a least square surface on the data of figure 4. A curve is fitted for each pixel coordinate independently. When observing the calibration parameters for all CCD pixel coordinates the contribution of the alignment errors, lens aberrations, scaling errors and rotation errors can be separated. Using the calibration parameters and the measured local phase gradient it is now possible to calculate the true thickness of a wafer independently of wafer tilt or bow. Figure 5 shows the construction of a thickness phasemap from an aligned measurement and from a strongly tilted measurement. The measured wafer thickness variation is obtained from both flatness phasemaps (A and B). After applying correction for the thickness reference (C) and a correction (D) for the mentioned error sources the true thickness (E) is obtained. Both calculated thickness maps show a good resemblance. A thickness phasemap of the entire

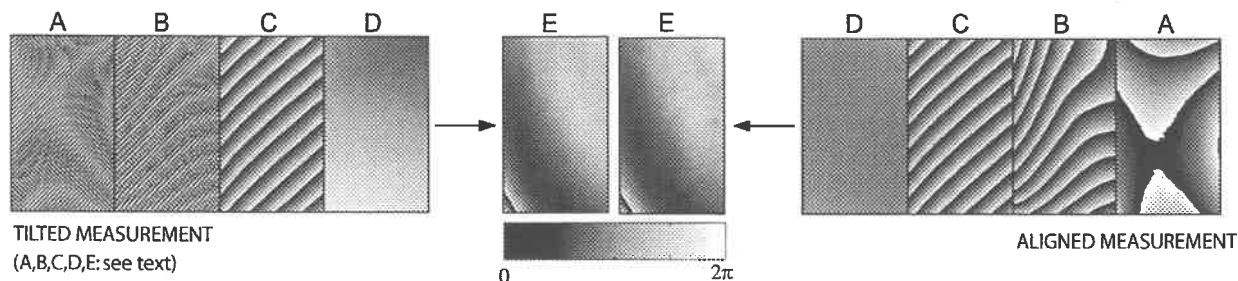


Figure 5. Left: Tilted measurement (carrier fringe method), Right: Aligned measurement (phase shifted method)

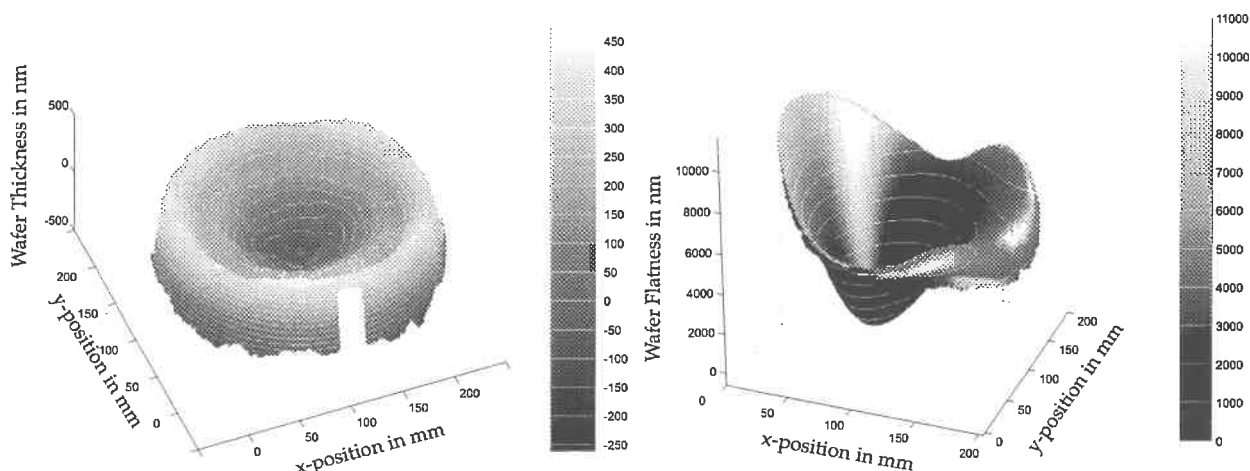


Figure 6. Left: Wafer thickness variation, Right: wafer flatness

wafer is constructed by combining all locally measured thickness maps. After unwrapping this phasemap the thickness variation of the entire wafer is obtained (figure 6). For combining all local flatness measurements into a total flatness map of the wafer a surface stitching software package has been developed^{1,3}

4. CONCLUSION

Thanks to proper calibration methods the thickness variation of a wafer can be measured by means of a double side scanning fizeau interferometer. The calibration method allows the use of relatively cheap optics and makes the thickness measurement insensitive for wafer tilt and bow.

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Roundness, angle, straightness and waviness measurements on recessed cones using scanning white-light interferometry

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1. Introduction

Valves are key components of the modern high-pressure fuel injection and hydraulic systems that power cars and trucks. The usual functional surfaces in a valve are the concave seat (a subregion of a conical surface) and a mating convex part (a subregion of a sphere or cone). Valve seat roundness tolerances are on the order of one micrometer or less, while cone angle tolerances are on the order of one degree or less. These tight tolerances require dedicated mechanical roundness profilers both on the factory floor and in the QC laboratory to validate the manufacturing process. Desirable improvements to valve metrology going forward include higher speed and lateral resolution, as well as challenging Gauge Repeatability and Reproducibility (GR&R) targets, all of which point towards optical solutions.¹

We report in this paper a new experimental configuration of a scanning white-light interferometer that allows measuring the critical parameters of recessed valve seats. Typical surface profilers based on interference microscopy provide surface form and roughness by comparing the surface of interest to a plane datum. This is however not the ideal configuration to measure a conical surface, since the direction of illumination and observation is not normal to the surface. On the other hand, optical systems capable of generating a conformal conical wavefront with a sufficient lateral resolution over a large spectral range are difficult to design and manufacture and are by nature dedicated to discrete cone angles. We investigate here a flexible optical geometry that allows measuring a wide range of cone angles at various inspection diameters with a single optical system.

2. Measurement concept

The key idea is an interferometer that uses a datum point in place of the usual plane datum surface.² The measurement process brings this datum point at a location such that it becomes the center of a virtual mating sphere contacting the valve seat at the nominal seat diameter. The interferometric data allows measuring the distance between the sphere and the actual surface. Because the sphere and cone are tangent at the nominal seat diameter the direction normal to the cone surface corresponds to the sub-nm height resolution of interferometric measurements, providing the best possible functional metrology for the critical measurement of roundness at the seat. Adjusting the radius of the virtual mating sphere allows in principle to adapt the interferometer to any cone angle and seat diameter.

Figure 1a shows a conceptual instrument design. The core is a white-light Linnik interferometer. The optics on each leg of the interferometer are identical. They map a spherical object surface onto an intermediate image plane, which is imaged onto a camera by a telecentric relay. Chief rays in object space cross at the center of the entrance pupil. This location is the so-called datum point, the center of the virtual mating sphere to which the cone surface is locally compared. Practically, dedicated optics have

been designed in order to fit inside an electroformed tube, as seen in Figure 1b, thus forming an endoscopic objective capable of reaching deeply recessed valve seats.

The reference leg optics include a concave spherical reference mirror that is concentric with the reference datum point. This defines a virtual spherical surface in object space where high-contrast fringes can be observed. A displacement of the reference leg optics over a few tens of μm forces the radius of this surface to change linearly, as if the spherical surface was inflating. Practically, interference data is recorded during such scans where the virtual sphere passes through the physical part surface. A large displacement of the reference leg optics is also used to offset the center of the scan, thus allowing to change the inspection diameter on a given part or to measure at the same inspection diameter for a part having a different included angle. In all cases, the region surrounding the nominal valve seat remains tangent to the virtual mating sphere. The introduction of large radius offsets may require refocusing the camera in some cases.

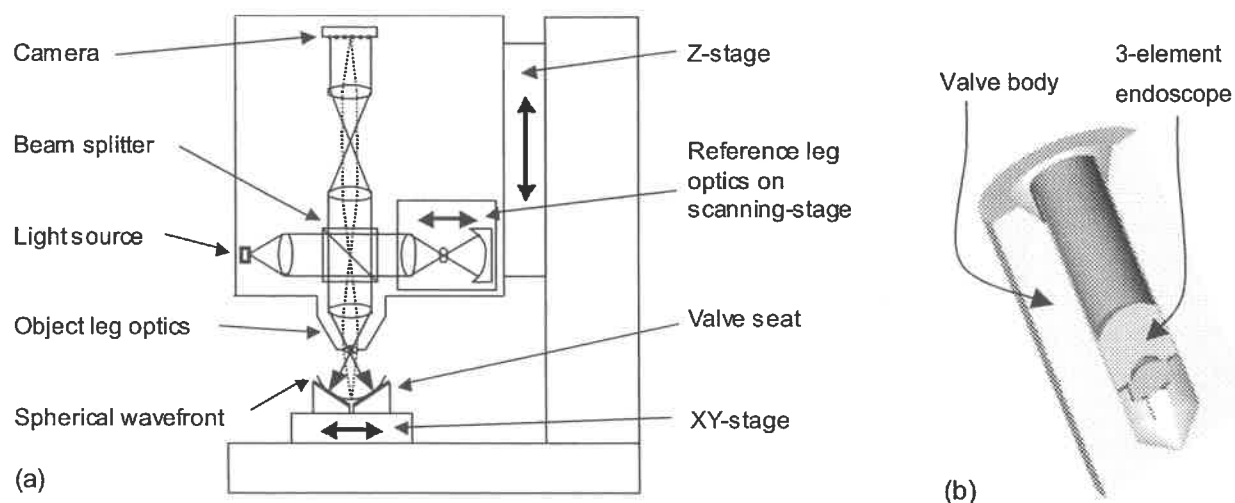


Figure 1 (a) Conceptual design of the optical gauge. (b) 3-element endoscope used both on the reference and object legs of the interferometer, shown here when measuring a deeply recessed valve seat.

3. Preliminary results of cone roundness and cone angle measurements

We studied the concepts outlined above on proof-of-concept breadboards. Data is captured during a scan of the reference leg. The raw height data obtained by processing of the interference data³ looks like the top of a donut (it is a map of the distance from the conical surface to a virtual mating sphere). As an example, the band of data captured at a 2.7 mm inspection diameter on a 90° cone is typically on the order of 0.3 mm wide along the cone generatrix for a typical production surface finish. These data are transformed into a 3D representation of the object surface according to the projection defined in Figure 2a. A point P on the object maps to a point P' on the detector. A calibration using cones of different included angles establishes the relationship between the observation angle θ and the radius ρ at the detector. Another calibration with a sphere of know radius combined with the interferometric measurement of the conical surface allows the measurement of the distance r from the point datum to the

object point P. Given these relationships a portion of the conical seat is reconstructed as shown in Figure 2b. A best-fit cone is then used to find the cone axis and included angle.

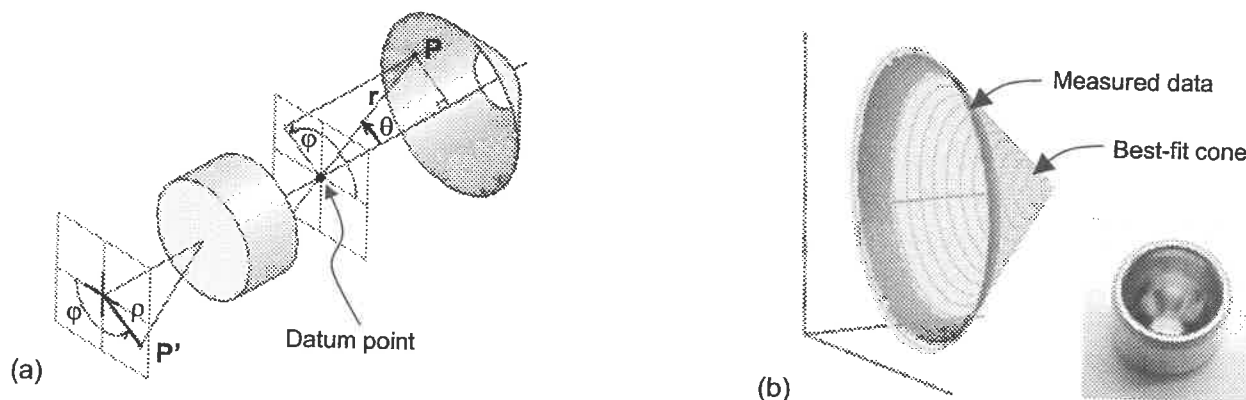


Figure 2 (a) Coordinate mapping of object point P onto detector point P'. (b) Reconstructed 3D surface and best-fit cone for a machined valve seat. A band of data about 0.35 mm wide is captured about the nominal seat inspection diameter.

Because the object surface is reconstructed in a coordinate system based on the datum point and the optical axis the best-fit surface indicates the amount of decentration and tip/tilt of the cone with respect to the interferometer. This information could be used on an automated instrument to readjust the cone position using the XYZ stages of Figure 1a if any of these parameters exceeds a predetermined threshold. This implies at most two measurements for a given part. By comparison, traditional tactile roundness profilers require a time-consuming iterative process for centering the part on the profiler spindle.

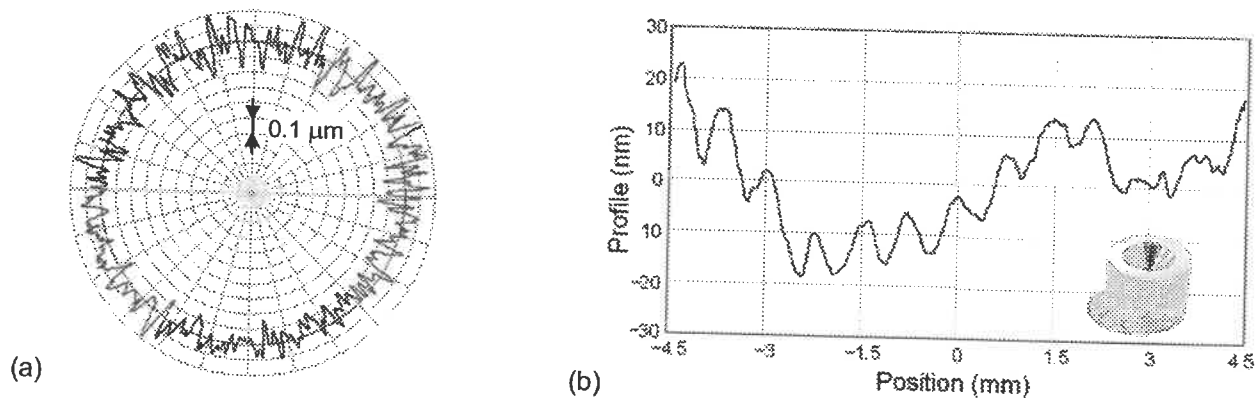


Figure 3 (a) Roundness profile at a specific seat diameter, extracted from the data in Figure 2b. (b) Roundness profile plot for a diamond-turned conical seat. Note the surface oscillation on the order of 10 nanometers caused by the spindle motor poles of the diamond turning machine.

Subtraction of the best-fit cone from the surface data allows calculating the surface deviation along the local cone normal. One can then extract a roundness deviation profile at a location corresponding to a specific seat diameter. Different types of filters can be applied to the roundness profile to extract form,

waviness and roughness. Additionally, profiles taken along the cone generatrix provide information about the lay pattern left by the machining process. An example roundness profile is shown in Figure 3a for the machined 90° valve seat shown in Figure 2b. The surface finish is fairly rough, as seen by the high-frequency surface deviation. The PV roundness deviation is about 0.4 μm for this particular part.

For calibration purposes we also manufactured diamond-turned conical surfaces. In this case the surface finish is fairly smooth compared to the mean wavelength of light. A roundness profile is shown in Figure 3b. The roundness error is smaller than 0.04 μm on this part. The resolution of the instrument also reveals a periodic 10-nm surface oscillation, which was found to correspond to the 12 poles of the motor driving the diamond-turning machine spindle.

A traditional metric in the automotive industry is the use of a GR&R study to validate the overall repeatability of a particular gauge. This typically consists in measuring a given set of parts multiple times, each part being loaded and unloaded between measurements. We ran such GR&R studies on conical seats having a surface finish similar to that of production parts and on smoother diamond-turned cones. The

results for cone roundness and included angle are summarized in Table 1. We report here the standard deviation corresponding to the gauge repeatability. Usual practice requires the range defined by $5.15 \times$ (standard deviation) to be smaller than 10% of the tolerance. The table shows that the performance of the proposed gauge is consistent with current submicron tolerances.

1- σ GR&R	Roundness	Included angle
Production-like cone	0.019 μm	0.012°
Diamond-turned cone	0.004 μm	0.001°

Table 1

4. Concluding remarks

The main benefits of the proposed gauge are those typically associated with optical profilers: non-contact collection of high-density 3D data instead of line traces (thus increasing the statistical significance of the data), as well as rapid data acquisition and repeatability/accuracy that compare favorably to that of mechanical profilers. Further work includes the study of alternative optical configurations,⁴ for example systems that attach to traditional interference microscopes.

5. Acknowledgments

We wish to thank P. Drabarek, M. Lindner, M. Kraus, R. Mason and C. McNeely of Bosch GmbH for their collaboration. D. Griffith, J. Everett, D. Imrie and S. Fantone of Optikos Corporation designed and manufactured two generations of advanced probes and additional optical systems for this project.

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Stabilization for Atomic Tracking Control of the Scanning Tunneling Microscope Tip by Referring Regular Crystalline Surface

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Introduction

Since nanotechnology and ultra-precision engineering have progressed very rapidly, a displacement measurement or a positioning control with a resolution of sub-nanometer or less is required. The scanning tunneling microscope (STM) ¹⁾ is a popular tool in surface engineering and can be used to obtain images of atoms on crystalline surface. The lattice spacing of approximately 0.2 nm for some regular crystalline lattices is uniform and stable over a long range when the crystals are stress free. Therefore, the crystals can be used as the references with a resolution of sub-nanometer or less for the displacement measurement and the positioning control by combining the STM ²⁾⁻³⁾.

A control method to position a STM tip onto an apex or bottom points of the sample of interest using lateral gradient signals ($\partial Z/\partial X$, $\partial Z/\partial Y$) was proposed by Pohl and Möller ⁴⁾, since the signals must be zero on the apex (or bottom) points. In the method, the lateral gradient signals can be obtained simultaneously using two-phase lock-in modulations of a tunneling current modulated with lateral-circular dither modulation applied to the XY motion of the STM tip, and the signals can also be fed back, to be zero, to a XY sample (or tip) scanner with a tracking controller so that the tip can reach an apex (or a bottom) points. Aketagawa et al., extended the method to an atomic region and demonstrated the atomic tracking and stepping control of the STM tip with atomic resolution by referring to atomic apex points and arrays on the graphite surface ⁵⁾. However, even if the STM is very stiff, outer disturbances (thermal drift, floor vibration and acoustic noise) can affect it so that its tip position jumps easily to some neighboring atoms or arrays in the case of a use of a simple proportional-integral (PI) method as the tracking controller. Since a lot of disturbances can affect the STM at daytime, it is very difficult to perform the atomic tracking control with the simple PI controller. Therefore, the high stabilization technique for tracking control of a STM tip by referring atomic points and arrays on crystalline surface must be developed. One of our aims is to stabilize and use the atomic tracking control even at daytime.

System Evaluation and Controller Design

Fig. 1 shows an outline of the STM instrument utilized for the atomic tracking ⁵⁾. The STM consists of a head (= a tip + a tip XY tube scanner) and a base (= a plate + a sample XYZ tube scanner + precision screws). Both the head and base are linked with stiff springs, which can be replaced to enhance the stiffness of the STM. To improve the performance of the tracking controller against the disturbance, an effect from the surrounding disturbance was evaluated by monitoring a power spectrum of the tunneling current without XY raster scanning. If the STM is enough stiff and no disturbance exists in the case of no XY raster scanning, no fluctua-

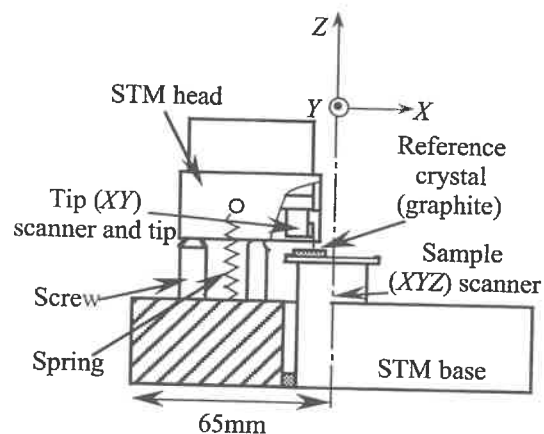


Fig. 1 STM head and base

tion of tunneling current in a power-spectrum can be observed. For the evaluation, we selected the following three environments, 1) daytime and normal stiffness (= default condition), 2) daytime and enhanced stiffness and 3) nighttime and normal stiffness. Normal lateral and vertical stiffnesses (resonant frequencies) of the STM are 2.8×10^6 N/m (340 Hz) and 2.4×10^6 N/m (320 Hz), respectively. For the enhancement of the stiffness, the number of the springs is doubled. Fig. 2 shows the outer disturbance effects on the power spectrums of the tunneling current. In the evaluation, the sample was the graphite crystal. The tunneling current and the bias were 2 nA

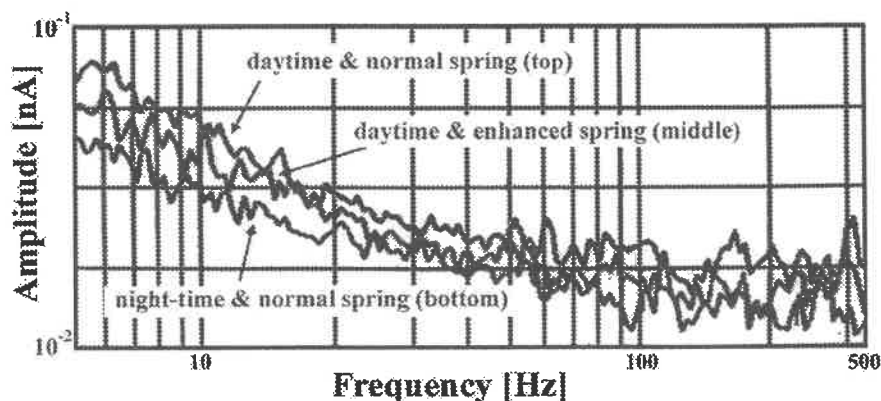


Fig. 2 Disturbance effects on power-spectrums of the tunneling current without *XY* raster scanning.

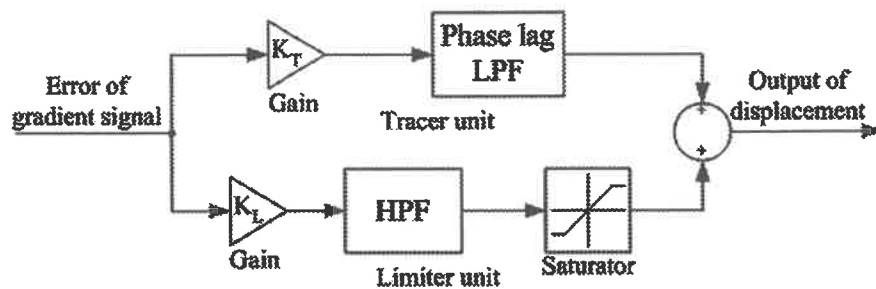


Fig. 3 New controller design.

and 50 mV, respectively. An anti-vibration table (resonant frequency = 2 Hz) was also used. From the Fig.2, we conclude that 1) an area whose frequency is lower than 100 Hz is a dominant outer disturbance at daytime because the area grows at daytime and 2) the enhancement of the STM stiffness is useful even at daytime. In the Fig. 2, tunneling current region whose frequency is larger than 200 Hz is almost white noise area.

Since a lock-in amplifier (LIA) in our control system, whose cutoff frequency is 160 Hz, limits a response of tracking control, the dominant disturbance at daytime can be compensated by designing an appropriate atomic tracking controller. It shows conversely that a disturbance whose frequency is over the cutoff frequency of the LIA must be suppressed by enhancing the STM stiffness. Fig. 3 shows a new controller design, which includes tracer and limiter control units. The tracer units is a phase lag low-pass filter (LPF) whose frequency response is flat widely, and can compensate a disturbance whose frequency is lower than 100 Hz by adjusting the cutoff frequency and the decay-slope. The limiter consists of a high-pass filter (HPF: cutoff frequency = 100 Hz) and a saturator. For a disturbance whose frequency is over 100 Hz, the limiter can restrict a displacement amplitude of its output within less than a half of the lattice spacing on the crystalline surface. The enhancement of the STM stiffness is also employed for suppressing a disturbance whose frequency is higher than 100 Hz.

Experiments

In order to assess the above stabilization technique (the new controller + the enhancement of the STM stiffness), a new experimental control system, which is shown in Fig. 4, was constructed. The system consists of the STM whose stiffness can be replaced, a current to voltage converter (I/V), a dither modulator, an artificial disturbance generator, a 2-phase LIA, the tracking controller, and the *Z*-servo controller. The artificial lateral disturbance is directly applied to the tip *XY* scanner from the generator. To achieve a flexible control, a DSP and 16 bit ADCs/DACs were utilized as the tracking controller. If the tracking controller can follow the artificial disturbance, a stabilization of the tracking control

can be realized. By varying the amplitude and the frequency of the artificial sinusoidal disturbance as experimental parameters, a critical controllable amplitude for a specific frequency were determined. Criteria for the stabilization are as follows;

- (1) no jump effect should occur during the control,
- (2) a displacement amplitude difference between the disturbance and the control signal should be less than 10 %,
- (3) a controllable duration should be longer than 1 min.

The experiments were made for both the new tracking controller with the enhanced STM stiffness and the simple PI controller, which was tuned manually, with the normal STM stiffness. In the experiments, the graphite crystal, whose lattice spacing is about 0.25 nm⁶⁾, was utilized as the reference. The tunneling current and the bias voltage were 2 nA and 50 mV, respectively. A sampling rate of the tracking controller was 5 kHz. Dither amplitude and frequency were 0.05 nm and 10 kHz, respectively. The experiments were made at nighttime to reduce the effects from outer disturbances, and the anti-vibration table was also utilized. Fig. 5 shows examples of artificial disturbances (upper) and control signals (lower). In the figure, left and right frames indicate performance

results of the new controller with enhanced spring and the simple PI controller with normal spring, respectively. In the figure, lateral and vertical axes are also time and displacement amplitude, respectively. Fig. 6 shows a relationship

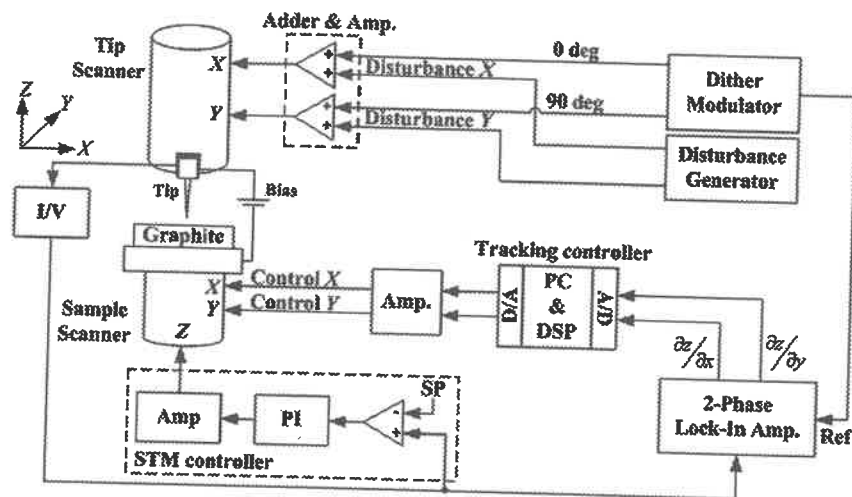
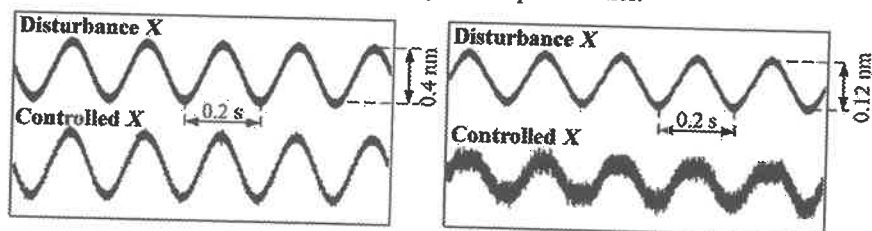


Fig. 4 Experimental control system. Artificial lateral disturbance is applied directly to sample scanner.



(a) The new tracking controller + enhanced springs. Disturbance amplitude and frequency are 0.4 nm and 5 Hz, respectively.

(b) The simple PI controller + normal springs. Disturbance amplitude and frequency are 0.12 nm and 5 Hz, respectively.

Fig. 5 Artificial disturbances and control signals from the controllers.

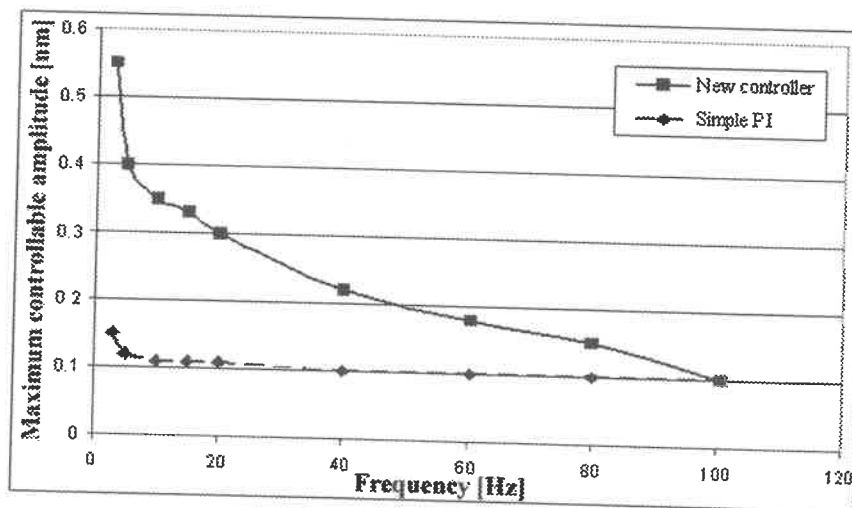


Fig. 6 shows a relationship between the maximum controllable amplitude and the frequency for the artificial disturbance.

between the maximum controllable amplitude and the frequency for the artificial disturbance. As shown in Fig. 6, the simple PI controller with the normal STM stiffness can only follow the disturbance whose amplitude is lower than a half of lattice spacing even in low frequency area. On the other hand, the new controller with the enhanced STM stiffness can even follow the disturbance whose amplitude is larger than the lattice spacing in a frequency area lower than about 40 Hz. Moreover, from the Fig. 5, a control quality of the new controller is better than one of the simple PI controller. The new controller with the enhanced stiffness can also allow the use of the atomic tracking control even at daytime condition. The results show the feasibility of the positioning control with atomic resolution using the stabilized atomic tracking control by referring atomic point and array on regular crystalline surface.

Conclusions

To improve the performance of the atomic tracking control against the outer disturbance, an effect from the surrounding disturbance was evaluated by monitoring a power spectrum of the tunneling current without a *XY* raster scanning. The dominant disturbance is located in the area whose frequency is lower than 100 Hz. The new tracking control algorithm and the enhancement of the STM stiffness were employed as the stabilization technique. The new control algorithm consists of tracer and limiter units. The tracer was designed to compensate the disturbance whose frequency is lower than 100 Hz. On the other hand, for the disturbance whose frequency is higher than 100 Hz, the limiter was designed to restrict a displacement amplitude of its output within less than a half of the lattice spacing on the crystalline surface. The enhancement of the STM stiffness is also employed for suppressing a disturbance whose frequency is higher than 100 Hz. To evaluate the stabilization technique, the artificial lateral disturbances were applied to the system. By varying the amplitude and the frequency of the artificial sinusoidal disturbance as experimental parameters, a maximum controllable amplitude for a specific frequency were determined, and the performance comparison between the new technique and the conventional PI controller with normal stiffness was made. The results show the new technique can compensate the disturbance of larger amplitude and higher frequency than the conventional method. The atomic tracking control can be made even at daytime with the new technique. The results show the feasibility of the positioning control with atomic resolution using the stabilized atomic tracking control by referring atomic point and array on regular crystalline surface.

Acknowledgments

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THIN GLASS OPTIC AND SILICON WAFER DEFORMATION AND KINEMATIC CONSTRAINT

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Measuring and assembling progressively thinner (length / thickness > 6) substrates is an increasing challenge. From disk drive substrates to flat panel displays, thin glass must be mechanically maneuvered without substantial distortion. Flatness of silicon wafers in the semiconductor industry is also becoming more important.

Grazing-incidence foil-optic x-ray telescopes (such as the segmented mirror approach considered for the NASA *Constellation-X* mission) require sub-micron accurate and repeatable assembly of thousands of foils with figure errors less than $0.5 \mu\text{m}$ over their $140 \times 100 \text{ mm}^2$ surface areas. To provide accurate metrology, a deep-UV Shack-Hartmann surface metrology system has been created [1].

To meet these assembly and metrology challenges, we present how thin materials such as silicon and glass wafers deform and how they can be constrained to minimize these effects. Both analytical calculations and finite element analyses (FEA) are utilized to understand the effects of gravity on foil deformation while varying parameters such as foil thickness and angle of inclination. Friction forces imparted during foil manipulation and thermal expansion mismatch between the foil and the constraint device are also studied.

These theoretical analyses lead to functional requirements for the design of a foil fixture—a device for holding these thin, floppy foils with kinematic mounting and minimal deformation. This air bearing device works in conjunction with the metrology system [1] to produce accurate and repeatable surface maps of the foil optics.

1. Modeling

1.1 Gravity Sag

For any simply-supported foil optic, gravity will typically cause distortions on the order of the thickness. Supported horizontally, the foil's maximum deformation will occur at its centerline as given by

$$\delta_{\max} = \frac{\rho g \sin \theta L^4}{6.4 E t^2}. \quad (1)$$

Using this equation, the deformations for 0.4 mm -thick, 140 mm -long silicon and glass foils are respectively $54 \mu\text{m}$ and $127 \mu\text{m}$. The $0.5 \mu\text{m}$ tolerance will not allow this manner of constraint. Orienting the foils vertically effectively reduces the gravity-induced load. For a hypothetical vertical misalignment of 2 mm at the top of a 140 mm foil, the resulting angle of inclination, or pitch, of 0.82° reduces the deformation to 0.77 and $1.81 \mu\text{m}$, respectively.

The three-dimensional deformation is more accurately modeled by FEA. This method yields a maximum displacement of $1.87 \mu\text{m}$, for glass with dimensions $140 \times 100 \times 0.4 \text{ mm}^3$, pitch of 0.82° , and pin boundary conditions, which compares well with the previous analytical beam bending result of $1.81 \mu\text{m}$.

1.2 Boundary Conditions

To accurately simulate actual foil distortion from gravity, we need to modify the boundary conditions. We have previously assumed pin joints along the top and bottom of the foil as in Figure 1a. To avoid overconstraint in practice, we should only contact the foil's edges at three locations as in Figure 1b. In general, this triad of ball-socket joints will allow more foil sag than the pin joint edges. An FEA simulation for glass with dimensions $140 \times 100 \times 0.4 \text{ mm}^3$ and pitch of 0.82° reveals a maximum deformation of $2.14 \mu\text{m}$ as compared with the previous result of $1.87 \mu\text{m}$.

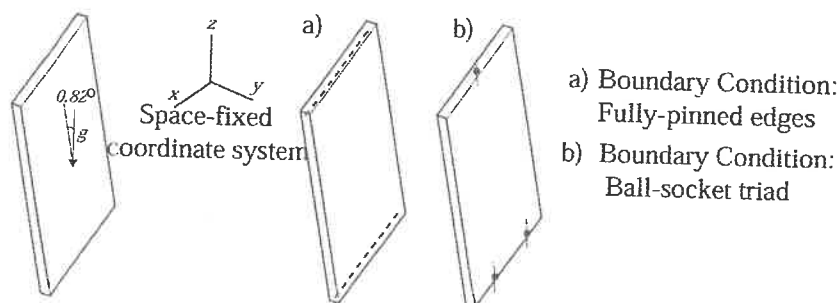


Figure 1. (a) Pin joint allows rotation about the x -axis only. (b) Ball-socket triad permits rotation about all axes.

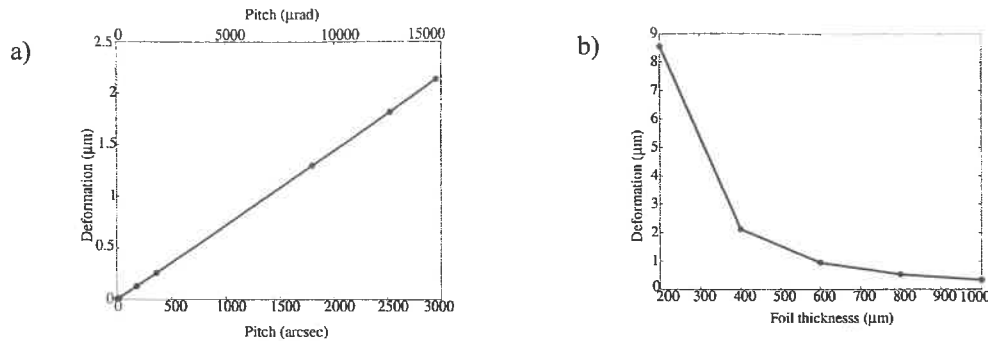


Figure 2. Maximum glass foil deformation as a function of (a) pitch angle for a thickness of 0.4 mm and (b) thickness for a pitch of 0.82°. Dimensions: 140x100 x0.4 mm³, Boundary conditions: ball-socket triad

1.3 Gravity sag as a function of pitch angle

Foil deformation as a function of pitch angle is shown in Figure 2a. These results help define the foil holder functional requirements. Specifically, if we allocate 15% of the allowable peak-valley (P-V) error (0.5 μm) to pitch angle repeatability, then this repeatability should be 100 arcsec at worst. The corresponding ~70 nm glass foil deformation should not significantly compromise assessment of the manufacturing quality.

1.4 Gravity sag as a function of foil thickness

The thickness of the foil affects how tolerant we can be of inclination errors. Equation 1 reveals that the deformation is inversely proportional to the thickness squared. Using FEA, Figure 2b shows the relationship. A 400 μm thickness was chosen for the telescope foils, balancing the needs for low mass and small deformations.

1.5 Thermal Considerations

Foil and holder expansion mismatch could inhibit accurate metrology. Linear thermal expansion is governed by

$$\Delta L = L\alpha_{th}\Delta T. \quad (2)$$

A hypothetical 7°C temperature change causes a 140 mm long aluminum fixture to expand by ~15 μm more than the glass foil inside of it. This difference distorts the foil if the holder forbids slip at the contact points (See Figure 3).

To estimate the reduction in foil warp from this expansion, the Figure 3 circle geometry can show that

$$\frac{2s}{c} = \frac{\theta}{\sin \frac{\theta}{2}} \quad (3)$$

Since $s=R\theta$, we can find R from the known s and c . Also from geometry,

$$h = R - R \cos \frac{\theta}{2} \quad (4)$$

Using $s=140$ mm and $c=139.985$ mm yields $h=888$ μm. This is much larger than the 0.5 μm tolerance. Thus, another functional requirement of the fixture design is that sliding between the foil and fixture must be permitted.

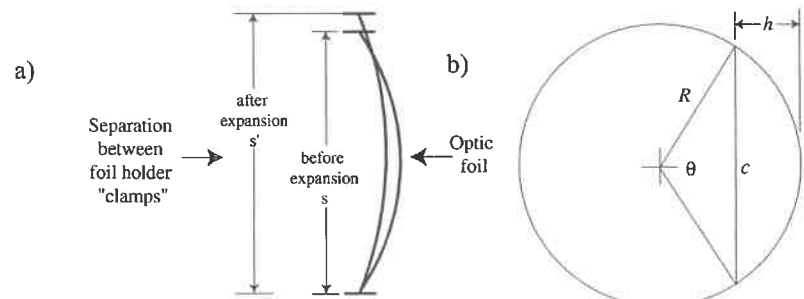


Figure 3. (a) Due to thermal expansion differences, the foil holder can reduce the foil warp if the boundary conditions do not allow slip. (b) The circle geometry is used to calculate this reduction.

1.6 Friction

Friction between foil and fixture can cause intolerable distortion. Consider an assembly scenario where a foil is slid into position by finger-like microstructure combs in accordance with other assembly research [3] (See Figure 4a). Comb teeth provide the actuation force, and friction arises from contact between the optic foil and comb base. Modeling as in Figure 4b, the friction force, F_{friction} is

$$F_{\text{friction}} = \mu mg \quad (5)$$

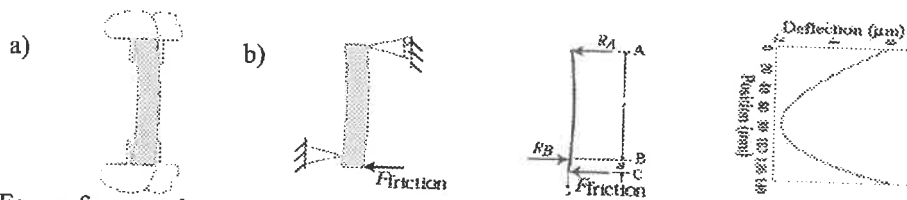


Figure 4. Forces from comb actuation and friction at the bottom of the foil can lead to distortion.

where μ , the coefficient of static friction, has been measured to be 0.39 [2]. The glass foil mass, m , is 14.1 g. The comb and foil dimensions are such that $a=1.09$ mm and $l=138.91$ mm. The maximum beam deflection is then

$$\delta_{\max} = \frac{F_{\text{friction}} a^2 (l+a)}{3EI} = 1.9 \mu\text{m} \quad (6)$$

An FEA simulation with the ball-socket triad gives $2.76 \mu\text{m}$, which compares well to the analytical result, indicating that friction can cause deformation beyond tolerances. A fixture design should therefore reduce or eliminate this friction. From these requirements, concept generation progressed to the design and fabrication of the foil fixture.

2. Experiment

The design of an optic foil holder with minimum friction forces is investigated. This holder utilizes two sets of air bearings, top and bottom, to kinematically constrain the foil's six degrees of freedom as shown in Figure 5.

2.1 Description

The top set consists of three pairs of circular, inherently-compensated, opposing air bearings to constrain translation along the x-axis and rotation about the y- and z-axes. Small air gaps between the bearing and the optic surface validates the use of viscous properties of air in describing the pressure within the bearing region. The viscous layer results in thrust due to the pressure distribution within this region as described by

$$p = p_1 \left[1 - \left\{ 1 - \left(\frac{p_{\text{atm}}}{p_1} \right)^2 \right\} \ln(r/r_1) / \ln(r_2/r_1) \right]^{1/2} \quad [4] \quad (7)$$

where p_{atm} is atmospheric pressure, p_1 is supply pressure, r_1 is inner radius, r_2 is outer radius, and r is distance from the center of the bearing as shown in Figure 6a. The load capacity of air bearings is proportional to the bearing area. This setup utilizes tubes with outer diameter of $635 \mu\text{m}$ to avoid gap variation associated with surface warp of the optics and thus reduce air bearing instability. Since the foils are held vertically, the load on the bearings is very small and the use of small diameter tubes is justified in the viscous region.

The bottom set consists of two inherently compensated, multiple inlet, circular bearings that constrain translation and rotation along the z- and x-axes respectively. These bearings have almost zero stiffness along the lateral direction, which allows them to adjust with lateral motion caused by thermal expansion mismatch and foil placement friction. The current design utilizes four feedholes with dimensions shown in Figure 6b. The load on these bearings is small, so vacuum preloading is used to increase stiffness.

2.2 Procedure

The top set is divided into reference and spring bearings. The reference bearings are stationary and initially use vacuum to hold the foils vertically. The spring bearings face the reference ones and are pneumatically actuated in the x-direction to facilitate insertion of optics. As the spring bearing approaches the optic, thrust forces increase. A

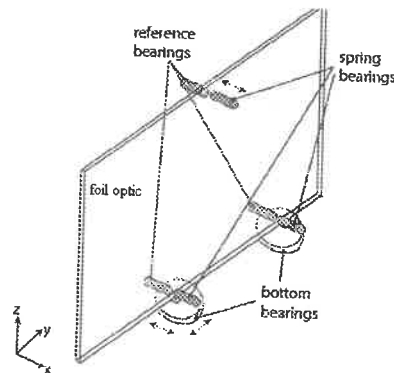


Figure 5. Two sets of air bearings, top and bottom, are used to kinematically constrain foil optics. The top set is divided into reference and spring bearings, which are exaggerated in size for clarity.

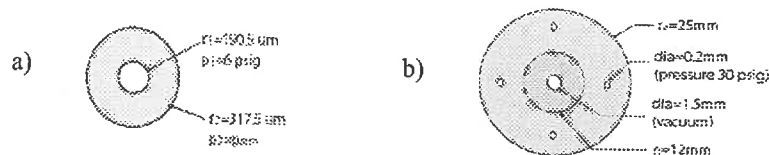


Figure 6. Geometry and dimensions of (a) top bearings and (b) bottom bearings.

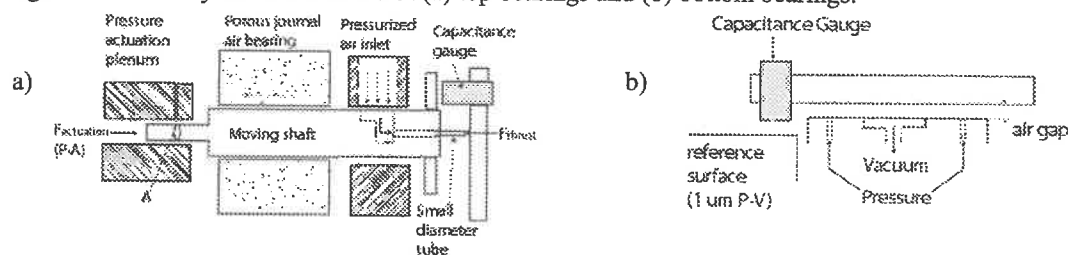


Figure 7. (a) Top bearing force versus gap calibration and (b) bottom bearing gap and stiffness vs vacuum calibration.

calibration curve is constructed for such bearings. A capacitance gauge is used to measure the gap between the air bearing and a conductive surface, and the force of actuation is equated to the thrust force in the absence of frictional forces, which are minimized by using a journal air bearing to carry the shaft of the top spring bearing as shown in Figure 7a. Contact between bearing and conducting surface is indicated by an electrical short circuit.

The actuating pressure is gradually decreased while the air gap is measured. Since the tube outer diameter is small, certain aspects must be altered to enhance the proper development of radial boundary layers that form the basis of the viscous flow between the two surfaces, such as the decrease in supply pressure and increase in tube inner diameter, to reduce the entrance effects of separation caused by the turning of the jet at the bearing exit. After proper positioning of the spring bearings, the reference bearings pneumatically center the foils.

The bottom bearing is supplied with air at 30 psig. Different vacuum pressures are used to study the stiffness of this configuration. A load of 55 g is applied to measure the decrease in air gap and calculate the stiffness. The gap is measured with a capacitance gauge as shown in Figure 7b.

2.3 Results

To ensure proper bearing performance, the flow of air must be within the viscous region, which can only be achieved at very small air gaps. Figure 8a shows that a gap of around 8 μm for the top bearing is sustained at pressures around 0.18 psig acting on a 2.54 mm diameter shaft. This calibrated curve depends on the spring bearing assembly orientation—pitch errors introduce the shaft weight, and therefore require a new calibration curve.

Figure 8b shows the effect of vacuum on the bottom bearing stiffness, which rises rapidly at lower vacuum levels and stabilizes at around 20 in Hg, providing a good operating region. The bearing has demonstrated some lateral instability due to the inherent property of negligible lateral stiffness of thrust bearings.

3. Conclusions

The behavior of thin optics under loads such as gravity, thermal expansion, and friction has been presented as modeled analytically and using FEA. From these studies, the requirements for the design of a foil holder have been derived, and the essential element in this foil holder, the constraint bearings, have been design and evaluated.

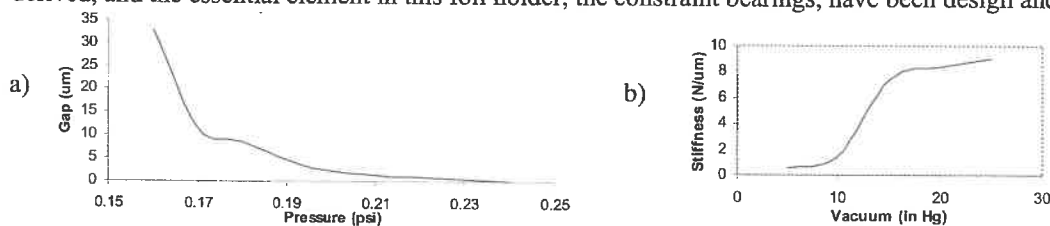


Figure 8. Plot of (a) air gap versus actuating pressure for top bearings and (b) stiffness versus vacuum for bottom bearings.

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DYNAMIC FRICTION COEFFICIENT MEASUREMENTS: DEVICE AND UNCERTAINTY ANALYSIS

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1.0 Introduction

Friction is an important consideration in the design and control of precision positioning systems. The detrimental effects of friction are widespread and include, for example, wear at sliding contacts, heat generation under dynamic conditions, stick-slip behavior, force discontinuities at changes in velocity direction, and limitations in start/stop resolution [1]. On the other hand, dynamic friction can also provide a source of damping. For these reasons, accurate friction coefficient values, defined as the dimensionless ratio of the friction force vector magnitude to the normal force vector magnitude, are required by designers for many material pairs.

Although efforts to better understand and model frictional behavior continue, laboratory experimentation remains the only practical method available for the accurate identification of friction coefficients for arbitrary material pairs. Even in the laboratory setting, however, accurate and repeatable friction coefficient measurement remains challenging due to the dependence of friction coefficients on the materials in contact, sliding speed, contact pressure, environment, cleanliness, normal load magnitude, surface topography, and many other factors.

Dynamic friction coefficient measurements are most often carried out using a tribometer, which applies a normal force to a pin that is pressed against a smooth, flat, rotating or reciprocating disk. The normal force may be applied with a dead weight load, or with a force actuator such as a pneumatic cylinder. The pin is held essentially stationary and a load cell monitors the sliding frictional force. The normal force is inferred from the dead weight or directly recorded using a load cell. The dynamic coefficient of friction is then calculated from the measured normal and friction forces.

In this paper, we present an uncertainty analysis, following the guidelines provided in references [2-3], for friction coefficient measurements carried out using a traditional pin-on-disk setup. This tribometer uses a pneumatic cylinder and a multi-channel load cell located directly above the pin to continuously monitor the contact force vector (see Fig. 1).

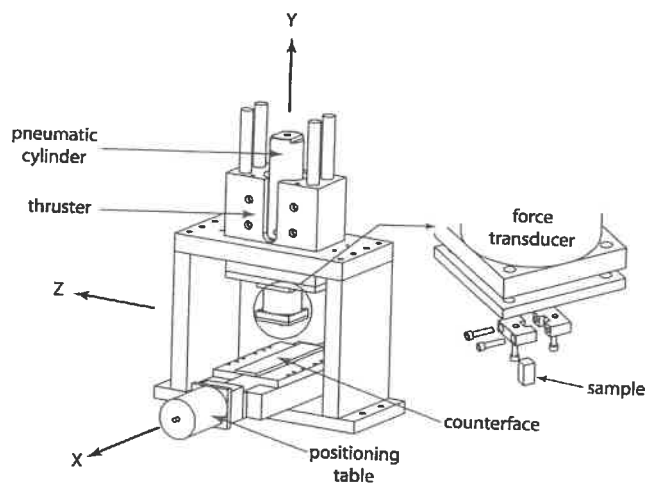


Figure 1 Schematic of the reciprocating tribometer constructed for this study

2.0 Description of Experiments

Twenty experiments were conducted using a polytetrafluoroethylene (PTFE) pin and polished 347 stainless steel counterface. Commercial PTFE rod stock was machined to produce 6.35 mm x 6.35 mm x 12.7 mm prismatic pin samples. These samples were mounted in the sample holder, and then machined flat on the contacting surface. The 347 stainless steel counterface was wet-sanded with 600 grit sandpaper, cleaned with soap and water, and wiped with acetone prior to each test. After preparation, all counterface surfaces were examined using a scanning white light interferometer to verify an average roughness (R_a) between 0.1 μm and 0.2 μm .

During testing, a 175 N normal force was maintained using the load cell's instantaneous force level as feedback to the pneumatic cylinder air supply (regulation was achieved using an inline electro-pneumatic valve). The average contact pressure for these tests was approximately 4.35 MPa. Tests were carried out for approximately 90 minutes, during which 2400 cycles were commanded over a 50.8 mm path length.

3.0 Friction Coefficient Calculations

The instantaneous coefficient of friction, μ , is defined as the ratio of the measured friction force, F_f , to the measured normal force, F_n , as shown in Eq. 1. In the most general case, the friction and normal forces at the contact are measured separately using some combination of force transducers and/or dead weight loads. Misalignments between the force measurement axes and the directions normal (N) and tangent (T) to the reciprocating or rotating surface constitute one of the most significant contributors to friction coefficient measurement errors. See Fig. 2.

$$\mu = \frac{F_f}{F_n} \quad (1) \quad F_x = F_f \cos \alpha + F_n \sin \alpha = F_n (\sin \alpha + \mu \cos \alpha) \quad (2) \quad \mu = \frac{F_x \cos \beta - F_y \sin \alpha}{F_y \cos \alpha + F_x \sin \beta} \quad (3)$$

$$F_y = F_n \cos \beta - F_f \sin \beta = F_n (\cos \beta - \mu \sin \beta)$$

Here, the normal and tangential directions are defined by the surface of the counterface. The intent of the instrument designer is to measure forces along these directions. However, manufacturing tolerances inevitably produce some misalignment between the axes of the force measurement system and the N-T axes. In Fig. 2, the transducer axis that is intended to measure the normal force is designated by Y and misaligned by β ; the transducer axis that is intended to measure the tangential force is designated by X and misaligned by α . In this general case, the measurement axes are not assumed to be perpendicular. The normal and tangential components of the contact force may be projected onto the measurement axes, resulting in the measured forces F_x and F_y , defined in Eq. 2. Using these projections, a solution for the friction coefficient in terms of the measured forces and misalignment angles can be derived; see Eq. 3.

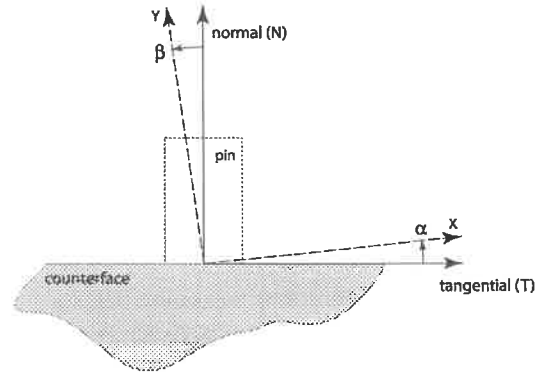


Figure 2 Force measurement geometry, where the misalignment angles are not necessarily equal

4.0 Uncertainty of Dynamic Friction Coefficient Measurements

In this case, the measurand μ is not observed directly, but is determined from Eq. 3. Prior to carrying out any uncertainty analysis, the known errors must be corrected or compensated, e.g., misalignment errors must be treated to avoid systematic errors in μ . However, even after all known error sources have been corrected/compensated, residual uncertainty remains due to the uncertainties in the measurements of the individual input quantities.

In order to evaluate the measurement uncertainty for μ , we apply the *law of propagation of uncertainty* to determine the combined standard uncertainty, u_c , which represents the estimated standard deviation (1- σ) of the measurement result. The combined standard uncertainty is a function of the standard uncertainty of each input measurement and the associated sensitivity coefficient, or partial derivative of the functional relationship between the friction coefficient and input quantity. These partials are evaluated at nominal values of the input quantities.

The expression for the square of u_c is provided in Eq. 4, where the standard uncertainty in each input variable, $u(x)$, can be determined using Type A or B uncertainty evaluations. In Type A, statistical methods are employed. Type B evaluations involve all other methods, which include using data supplied with a particular measurement transducer or relying on engineering judgment. It should be noted that Eq. 4 does not contain the potential for correlation (or dependence) between the separate input variables; zero covariance has been assumed in this analysis, as is often the case. The partial derivatives of the friction coefficient with respect to the input variables are also provided.

$$u_c^2(\mu) = \left(\frac{\partial \mu}{\partial F_x} \right)^2 u^2(F_x) + \left(\frac{\partial \mu}{\partial F_y} \right)^2 u^2(F_y) + \left(\frac{\partial \mu}{\partial \alpha} \right)^2 u^2(\alpha) + \left(\frac{\partial \mu}{\partial \beta} \right)^2 u^2(\beta) \quad (4)$$

$$\frac{\partial \mu}{\partial F_x} = \frac{\cos(\beta)}{F_y \cos(\alpha) + F_x \sin(\beta)} - \frac{\sin(\beta)[F_x \cos(\beta) - F_y \sin(\alpha)]}{[F_y \cos(\alpha) + F_x \sin(\beta)]^2}, \quad \frac{\partial \mu}{\partial F_y} = \frac{-\sin(\alpha)}{F_y \cos(\alpha) + F_x \sin(\beta)} - \frac{\cos(\alpha)[F_x \cos(\beta) - F_y \sin(\alpha)]}{[F_y \cos(\alpha) + F_x \sin(\beta)]^2}$$

$$\frac{\partial \mu}{\partial \alpha} = \frac{-F_y \cos(\alpha)}{F_y \cos(\alpha) + F_x \sin(\beta)} + \frac{F_y \sin(\alpha)[F_x \cos(\beta) - F_y \sin(\alpha)]}{[F_y \cos(\alpha) + F_x \sin(\beta)]^2}, \quad \frac{\partial \mu}{\partial \beta} = \frac{-F_x \sin(\beta)}{F_y \cos(\alpha) + F_x \sin(\beta)} - \frac{F_x \cos(\beta)[F_x \cos(\beta) - F_y \sin(\alpha)]}{[F_y \cos(\alpha) + F_x \sin(\beta)]^2}$$

4.1 Force Calibration Standard Uncertainty

A multi-axis force transducer, capable of measuring forces in three nominally orthogonal axes as well as moments about these axes, was used to measure the contact force. The force transducer had a maximum load capacity of 2200 N along the Y axis (vertical direction) and an 1100 N capacity in the X and Z axes. The transducer and conditioner pair output a voltage, V_i , which was then multiplied by a calibration constant, C_i , to obtain the desired force values.

In general, crosstalk exists between the force transducer axes. Thus, the voltage output for the axis in question is predominantly due to the load applied to that axis, but is also slightly dependent on loads applied to the other axes. The crosstalk matrix supplied by the manufacturer was used to compensate the measurement data. However, in performing the uncertainty analysis, we were not interested in the magnitude of bias corrections, but rather in the uncertainty of the corrections. To estimate the uncertainty of the crosstalk corrections, the uncertainty of each term in the crosstalk matrix must be known. For the transducer used in these experiments, the crosstalk between any pair of axes was 2% or less according to the manufacturer. It was assumed that the uncertainty in the cross talk was much less than the nominal value; therefore, the uncertainty of the crosstalk compensation was neglected in this analysis.

The measured friction and normal forces, F_X and F_Y , respectively, were calculated from the product of the recorded voltages V_X and V_Y and the corresponding calibration constants, C_X and C_Y . The X and Y direction calibration constants for the force transducer were determined by applying known dead weight loads to the transducer axis of interest and recording the corresponding voltage. The slope of the applied force versus voltage trace was then taken to be the calibration constant for that axis.

The uncertainty in the measured calibration constants was dependent on uncertainties in both the measured voltages and the forces applied during calibration. The voltages were recorded using a 12-bit card with an operating range of 20 V. The resulting quantization error (+/- least significant bit), ΔV , was 4.9 mV. The dead weight loads were measured on a digital scale with an uncertainty of 0.025 N. Because there are uncertainties in both voltage and load, a Monte Carlo simulation was completed to generate calibration curves based on both the mean dead weight values and corresponding uncertainty and the mean recorded voltages and uncertainty (normal distributions were assumed). Using the simulation, 10000 calibration constants were generated. The mean and standard deviation of the simulation results were taken to be the calibration constant and standard uncertainty, respectively, for each axis (148.77 N/V and 0.39 N/V for the Y direction; 35.99 N/V and 0.11 N/V for the X direction).

4.2 Voltage Measurement Standard Uncertainty

During experiments, the X and Y direction transducer voltages were subject to Gaussian noise. Therefore, the friction coefficient was computed using time-averaged transducer voltage measurements. The uncertainty in the average measured voltages was obtained using Eqs. 5 and 6, where $s^2(V_X)$ and $s^2(V_Y)$ are the variances in the voltages recorded by the force transducer in the X and Y directions, respectively, n is the number of samples, and $\bar{V}_X$ and $\bar{V}_Y$ are the mean recorded voltages in the X and Y directions, respectively [4].

$$u^2(\bar{V}_X) = \frac{s^2(V_X)}{n} = \frac{1}{n-1} \sum_{i=1}^n (V_{X,i} - \bar{V}_X)^2 \quad (5)$$

$$u^2(\bar{V}_Y) = \frac{s^2(V_Y)}{n} = \frac{1}{n-1} \sum_{i=1}^n (V_{Y,i} - \bar{V}_Y)^2 \quad (6)$$

For typical data, the mean voltage values were 0.893 V for the X direction and 1.336 V for the Y direction. This data was collected near the mid-point of the reciprocating path stroke. The standard uncertainties in these average values were 1.53×10^{-6} V and 6.07×10^{-7} V for the X and Y directions, respectively.

4.3 Standard Uncertainty of Forces

As previously discussed, average X and Y direction force values, $\bar{F}_X$ and $\bar{F}_Y$, were used in the friction coefficient calculations due to the normally distributed noise in the experimental data. These forces were calculated using Eq. 7. The standard uncertainties in the mean values $\bar{F}_X$ and $\bar{F}_Y$ are given by Eqs. 8 and 9, respectively.

$$\bar{F}_X = \frac{C_X}{n} \sum_{i=1}^n V_{X,i} = C_X \bar{V}_X, \quad \bar{F}_Y = \frac{C_Y}{n} \sum_{i=1}^n V_{Y,i} = C_Y \bar{V}_Y \quad (7)$$

$$u^2(\bar{F}_x) = (\bar{V}_x)^2 u^2(C_x) + (C_x)^2 u^2(\bar{V}_x) \quad (8) \quad u^2(\bar{F}_y) = (\bar{V}_y)^2 u^2(C_y) + (C_y)^2 u^2(\bar{V}_y) \quad (9)$$

The calibration constant uncertainties (Section 4.1) and the average voltages measured during the friction coefficient testing (Section 4.2) were substituted in Eqs. 8 and 9 to obtain the uncertainties in the average force values. For the test configuration and procedure described here $u(\bar{F}_x) = 0.095$ N and $u(\bar{F}_y) = 0.541$ N.

4.4 Force Transducer Misalignment Uncertainty

The angular misalignment was determined by measuring the horizontal force produced by the application of a known vertical force. Residual force that remained after compensation for the manufacturer-specified crosstalk was considered to be the result of misalignment between the transducer axes and N-T coordinate system defined in Fig. 2. The mean angle of misalignment was found to be 2.45 deg from multiple repetitions using three different normal loads of 47.2 N, 92.6 N, and 138.2 N. The mean and standard deviation values, 2.45 deg and 0.02 deg, respectively, from this sample distribution were assumed to provide the best estimates for the parent population mean and standard deviation so that $u(\alpha) = 0.02$ deg. Further, it was assumed that the transducer axes were perpendicular, such that $u(\alpha) = u(\beta) = 0.02$ deg.

5.0 Combined Standard Uncertainty

The combined standard uncertainty in the friction coefficient was determined using Eq. 4, where F_x and F_y were replaced by $\bar{F}_x$ and $\bar{F}_y$, respectively. The partial derivatives (provided in Eq. 4) were evaluated at the nominal operating conditions to obtain the sensitivity coefficients. Substitution yielded a combined standard uncertainty of 8.3×10^{-4} . The nominal friction coefficient value computed from Eq. 3 was 0.127. Thus, the standard uncertainty was approximately 0.7% of the nominal value. The expanded uncertainty, U , for a >99% confidence level ($U = 3u_c$) was 24.93×10^{-4} , or ~2% of the nominal value. The reader may note that if the friction coefficient were calculated using only the ratio of the average forces ($\mu' = \bar{F}_x / \bar{F}_y$) the resulting steady-state value would be 0.171. In this case, an uncorrected angular misalignment bias would produce a >34% error.

Instantaneous friction coefficient values for the 20 repeated tests are shown in Fig. 3. While variation in the data is apparent, not all the scatter can be attributed to the apparatus-based uncertainty described here. In fact, a significant source of variation in these measurements is the large number of factors that influence friction and the difficulty in controlling all of these variables. For example, it is impossible to use exactly the same PTFE sample for all tests because it is destroyed during testing. Also, the counterfaces must be repolished after each experiment. While every effort was made to carry out identical tests, no experimental artifacts exist that can be repeatedly measured under identical conditions to validate the analysis provided here.

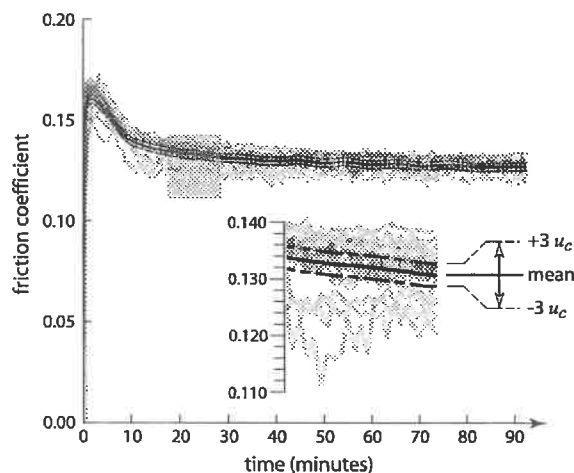


Figure 3 Friction coefficient data

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DESIGN OF DETACHABLE PRECISION FIXTURES WHICH UTILIZE HARD AND LUBRICANT COATINGS TO MITIGATE WEAR AND REDUCE FRICTION HYSTERESIS

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Keywords: Kinematic coupling, detachable, fixture, precision, alignment, hard coating, friction, repeatability, wear, Hertzian contact

1. INTRODUCTION

Detachable precision fixtures have reached a performance barrier defined by tribological phenomena driven by friction and wear. Although detachable fixtures with nanometer-level precision have been made using exotic and expensive materials [1], a low-cost means of achieving the same level of performance remains elusive. The goal of this paper is to introduce experimental results that indicate low to moderate cost hard coating of balls and grooves can improve the repeatability of detachable fixtures.

Coatings such as TiN have long been used to reduce friction and wear of high-performance tooling. These coatings can easily be adapted to do the same for the interfaces of detachable fixtures. Increasing the hardness of contacting surfaces, increases the energy required to change the surface (i.e. wear) [2]. Lower-friction coatings minimize the energy stored at the contacting interfaces [3], thereby reducing friction hysteresis and reducing the energy available to promote surface wear. This is important given the high energy density and energy density gradients near-the point contacts of balls and grooves in kinematic couplings.

Based upon the preceding knowledge, a "first generation" experiment was run to indicate whether the hypothesis that hard coatings improve repeatability may be true. The experiments and results are discussed in the following paper. These efforts form the basis for a more detailed, "second generation" of experiments which will be used to create deterministic models that link coating characteristics and properties to coupling repeatability.

2. EXPERIMENTAL PROCEDURE AND SETUP

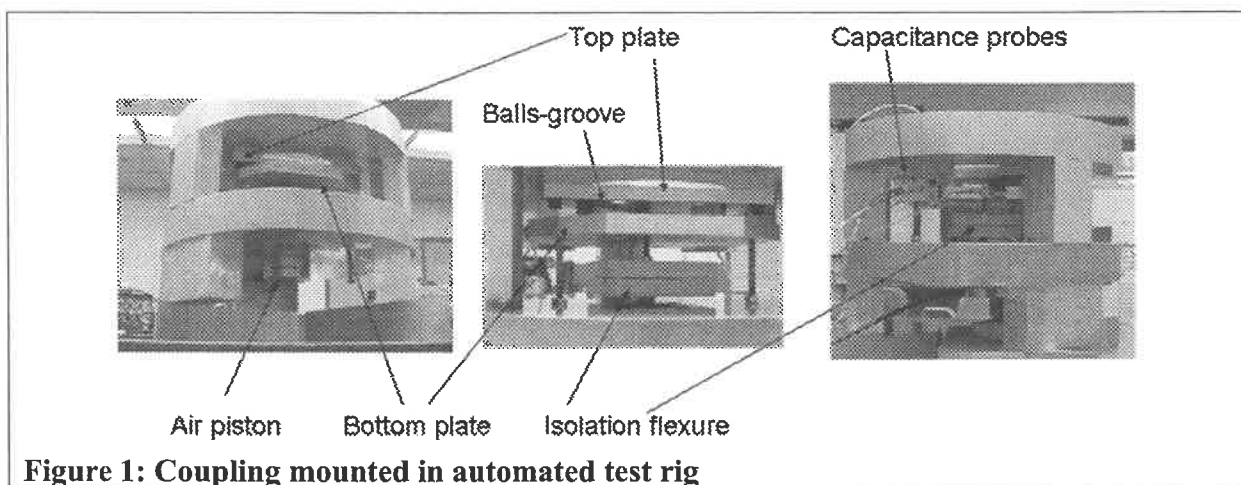
Three identical sets of hardened stainless steel kinematic elements (balls and grooves) were prepared for testing (ground surfaces). The first set serves as a control and thus the surfaces of the balls and grooves are not treated. The second set was treated with TiN coating. The third set was treated with TiCN. Characteristics of the surfaces of each set are provided in Table 1.

Table 1: Characteristics of ball-groove surfaces without lubrication

	Hardness RC	Coefficient of Friction	Coating Thickness
440 Stainless Steel	50 - 55.	0.80 (varies by 50%)	N/A
TiN	85	0.50	3.5 microns +/- 1.5 microns
TiCN	86 - 90	0.40	3.5 microns +/- 1.5 microns

Figure 1 shows a test kinematic coupling set mounted within an automated test rig. This test rig

is equipped with an automated air piston for engaging, disengaging and preloading the coupling set with 450 N of load. Lateral piston forces (those which cause error in coupling plane) are decoupled from the coupling via an isolation flexure place between the coupling and the piston. Six capacitance probes are positioned to read the displacement of the bottom plate (balls) with respect to the top plate (balls). The top plate and capacitance probes are rigidly attached to the test rig frame. A DSpaceTM interface is used to control the air piston (binary on-off coupling preload) and to read data from the capacitance probes.



The control/data acquisition is programmed to engage the coupling, then allow the coupling components to settle with respect to each other for 30 seconds before data is taken. This procedure is followed to achieve a stable position equilibrium that occurs after transient settling motions cease. In these tests, the transients could not be sensed after 25 seconds. Once the coupling is settled, 100 data points are taken within one second to average out noise and outliers in the readings. The coupling is then disengaged and the cycle repeats.

3. TEST RESULTS

Test results have been collated in Table 2. Figure 2 shows raw plots of displacements from each cap probe in an example repeatability test. This data contains approximately 15nm noise from instrumentation and other error due to preload variation caused by drift in the pressure supplied to the air piston. This drift is systematic and has thus been calibrated out of the data to obtain the readings provided in Table 2. The preload/pneumatic system has been replaced with a steady pressure supply for the next generation of experiments.

Table 2: Results of repeatability* tests [filtered** data]		
Ball-Groove surface	Linear [nanometers]	Angular [micro-radians]
Stainless-Stainless	332	3.7
TiN – TiN	179	0.3
TiCN - TiCN	34 [noise threshold = 15 nm]	1.0

* Preload drift error and pressure spike outliers have been removed to obtain this data

** Repeatability is calculated as $\frac{1}{2}$ (Range) with outliers removed

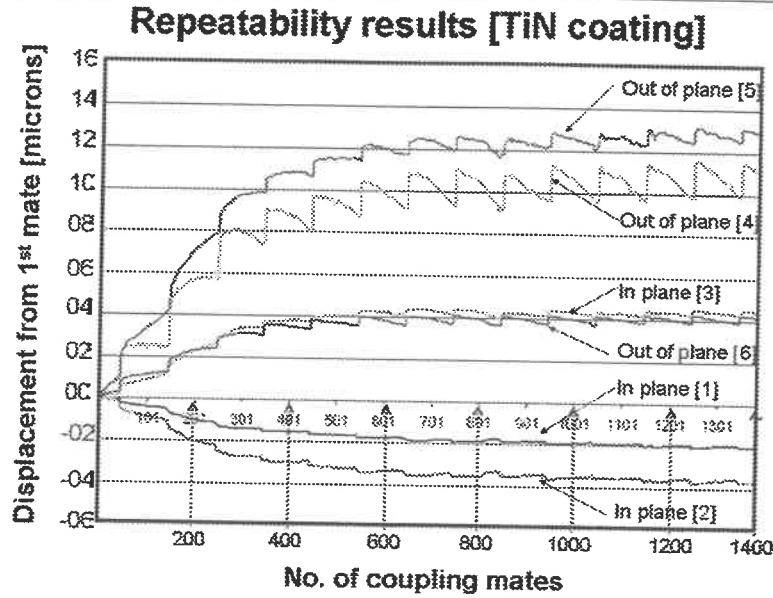


Figure 2: Example data from TiN coated repeatability test

Probe numbers [#] and reading direction (in or out of coupling plane) are indicated for each data plot

4. DISCUSSION

The intent of this paper has been to determine if hard coatings can have a positive effect on coupling repeatability. The experimental data provides a “first generation” level of proof that this is true. TiN coated performance is nearly a factor of two better than non-coated balls-grooves while TiCN performance may be factor of ten or better than non-coated balls-grooves. We use the words “may be” and “better” when describing TiCN performance as the measured/filtered repeatability is near the noise threshold of the measurement system. The full reasons for the large performance difference between the TiN and TiCN is not clear. It is likely that the difference is due in part to the higher hardness and lower friction coefficient of the TiCN (see Table 3). The fact that there is a larger performance difference between TiN and TiCN (moderate difference in characteristics) than there is between TiN and Stainless (large difference in characteristics) may indicate a non-linear relationship between coating characteristics and coupling performance. This of course may also be due to the inherent variation within the experiment.

Table 3: Results of repeatability* tests [filtered data]**

Ball-Groove surface	Linear [nanometers]	Angular [micro-radians]	Normalized Hardness	Normalized Friction Coeff.
Stainless-Stainless	332	3.7	1.0	1.0
TiN – TiN	179	0.3	1.6	0.6
TiCN – TiCN	34	1.0	1.7	0.5

These results warrant a second, more detailed experiment with better resolution and control of input variables. Results of this “second generation” of experiments will be discussed in October during oral presentation.

The trends of the data also indicate that coupling elements with non-lubricated hard coatings can wear in to a stable position, unlike non-coated couplings [4]. It is difficult to extract more information from the data obtained in these experiments as the driving, phenomena behind coupling wear and repeatability depend on characteristics with inherent variability (e.g. friction and surface finish effects). Subsequent research efforts are focused on acquiring data in experiments in which these sources of variation are more tightly controlled or measured. This data is being used to link coating characteristics and variations in coating characteristics to coupling performance. These efforts and the following additional results will be discussed at the oral presentation of this work:

- (1) Analyses to develop first order estimates which provide design guidelines and a means to quantify performance improvements versus cost of implementation.
- (2) Data which supports the hypothesis that the repeatability of the coupling depends upon the order of engagement of the ball-groove point contacts.
- (3) Implications for palletized manufacturing processes
- (4) Overview of continuing efforts to improve detachable fixtures

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A Laser Speckle Sensor for Compound Rotary-linear Motion Metrology

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Keywords: precision metrology, laser speckle, speckle pattern, image registration, feedback control, drift.

1 Introduction

In previous work [1], our group implemented a tilted-mirror interferometric sensor that can measure rotation of a hybrid rotary-linear axis as part of a 5-DOF machine tool intended for fabricating centimeter-scale parts. Attached to the shaft of the hybrid axis is a tilted mirror rotary sensor, which uses two laser interferometers to measure the orientation of a tilted 3-inch diameter mirror attached to the shaft. The sensor has a resolution of 1,336,000 counts/rev ($4.6 \mu\text{rad}$) corresponding to a linear resolution of $0.046 \mu\text{m}$ at a 1cm radius. However, the tilted-mirror's inertia dominates the rest of the axis, limiting accelerations, and its mass causes a cantilever resonance, limiting the controller bandwidth.

A promising alternative sensor that can be used for cylindrical motion metrology is a laser speckle based machine vision sensor. We have implemented such a prototype metrology system for a 4-inch diameter air bearing spindle. Off-line measurements from the experiments performed on our setup show that images obtained from the spindle surface can be used to measure angular displacements of the spindle with sub-pixel resolution. Real-time implementation on our prototype setup is expected to be able to run at update rates of 10 Hz, thus allowing for closed loop control with a bandwidth of the order of 1 Hz. Further, we predict that the sensor can achieve resolutions of $0.1 \mu\text{m}$ for translation and $5 \mu\text{rad}$ for rotation.

Yamaguchi [2] developed a laser speckle rotary encoder achieving a resolution of $35 \mu\text{rad}$ for angle of a 115-mm diameter metal cylinder. However, the encoder is based on measuring relative displacement across images. Measurements based on relative registrations are prone to significant drift, which cannot be tolerated for high precision measurements. In our implementation, we intend to eliminate drift by referencing all our image registrations to global images retrieved from a library generated over the range of motion.

2 Design of the Laser Speckle Sensor

Figure 1 shows the schematic of our prototype laser speckle based sensor. A laser beam is directed at the cylindrical surface of a rotating shaft. Parts of the reflected beam interfere in the space surrounding the illuminated spot on the spindle surface, creating a three-dimensional pattern, referred to as a speckle pattern [3].

An analog monochrome CCD camera captures, at a frame rate of 30 Hz, a two-dimensional slice of the pattern (see Figure 2) obtained at a fixed point in this space. When the surface is displaced laterally, the speckle pattern is found to shift accordingly, while slowly changing in the distribution of spatial intensity. Laser speckle can therefore be viewed as a way to recover displacements of surfaces from displacements across images.

A video frame grabber interfaced with the camera feeds the images to PC-based image registration algorithms that recover angular displacements of the spindle from the offset between images. We perform preliminary experiments with the sensor over small displacements given by rotating a micrometer head mounted on a cantilever arm fixed to the spindle as shown in Figure 3. We compare our measurements with those obtained from an auxiliary sensor consisting of an incremental optical rotary encoder indexed by a custom-built photo-interrupter unit mounted on the spindle.

To implement the sensor for feedback control of position, the video frame grabber is interfaced with a real-time operating system with an embedded real-time engine that implements the image registration algorithms. We provide low torque inputs to the spindle from a brushless DC motor via a belt drive. An optical encoder mounted on the back of the motor is used to measure angle and the difference between its successive counts is used to determine velocity.

3 Results

Off-line measurements obtained for small angular displacements of the spindle conform well with those obtained from the auxiliary sensor. Figure 4 shows on the 640×480 pixel plane the locus of the center of a pattern built from a reference image and detected across the images obtained when spindle is given small displacements with the micrometer totalling to $50 \mu\text{m}$ displacement along its surface. The repeatability of our measurement is within a pixel. The straight line locus lies along the direction of the speckle shift.

For large displacements of the surface, the speckle patterns de-correlate. To tackle this, we initially store a global library of reference images spread uniformly throughout the range of motion of the surface. Given the current image, we perform the image registrations with respect to the reference images retrieved from the library. The added advantage of this method is that we eliminate drift in our measurements, since our registration is based on global images. In our implementation, the library images are spaced at a 20-100 μm pitch. Further, we refine the estimate of the current position of the surface by averaging the results of image registrations performed on the current image with respect to the nearest neighboring reference images in the global library.

We have implemented closed loop position control of the air bearing spindle using feedback of position and velocity of the shaft of the brushless DC motor obtained from the optical encoder. The minor loops on the motor for velocity and position feedback from the encoder are constructed in dSPACE, a PC-based rapid prototyping controller, for cross-over frequencies of 150 Hz and 20 Hz respectively.

Further, we have implemented the feedback of angle estimates given by our laser speckle based sensor. We have observed that the jitter in the loop times for the image registration is bounded at 4.85 ms for images spaced 5 μm apart, thus allowing for real-time determinism. We expect the sensor to run at update rates of 10 Hz, thus enabling it for use in closed loop position control for bandwidths of the order of 1 Hz.

4 Conclusion

We have experimentally verified that the image sensor can be used to achieve sub-pixel resolutions. The measurements conform well with those obtained from an auxiliary sensor. With the existing setup, we predict that the sensor can achieve resolutions of 0.1 μm for translational and 5 μrad for rotational motion. However, the limitation of this metrology technique is the time and memory intensive nature of the image registration algorithms. By using localized search, we should be able to achieve higher update rates of around 30 Hz. We plan to continue to improve on the sensor and further, develop a similar version for measuring the rotation and translation of a hybrid rotary-linear axis.

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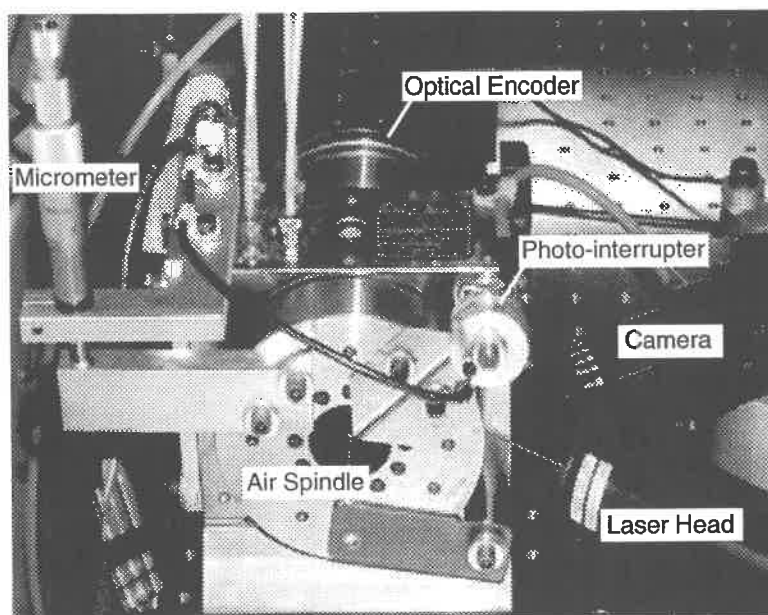


Figure 3: A laser speckle sensor for motion metrology of a precision air spindle. An optical rotary ring incremental encoder indexed by a custom-built photo-interrupter unit is used to compare results obtained from the speckle sensor.

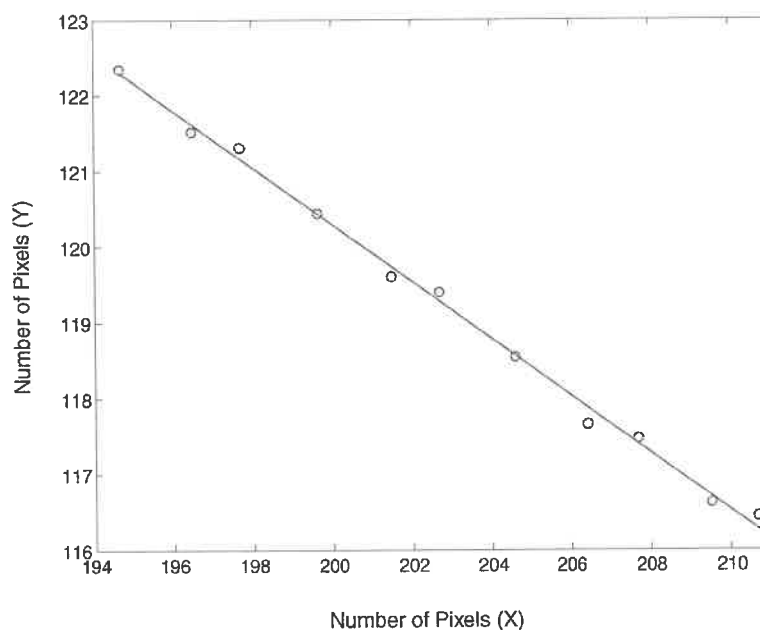


Figure 4: Line of motion of the centers of a pattern window matched across the images obtained for small rotations corresponding to a $50\text{ }\mu\text{m}$ displacement along the surface of the spindle.

Developing Atomic-resolution Measurements using a Tunable Diode Laser-based Interferometer

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Abstract

A new implementation of a Michelson interferometer capable of resolution on the order of 20 pm has been developed. This new method uses a tunable diode laser as the light source with the diode laser wavelength continuously tuned to fix the number of fringes in the measured optical path. The diode laser frequency is measured by beating against a reference laser. High speed, accurate frequency measurements of the beat frequency signal enables the diode laser wavelength to be measured with nominally 20 pm resolution. The new interferometer design is light weight and has been mounted directly on an ultra-high vacuum scanning tunneling microscope capable of atomic resolution. Most current commercial interferometry techniques are limited in resolution or accuracy due to polarization mixing errors, fringe interpolation limitations and errors, and environmental instability making them unsuitable or very difficult to use for the current measurement applications¹. In this paper, the new tunable diode laser Michelson interferometer described here addresses this class of measurements and calibrations.

Tunable diode laser-based Michelson interferometer principles

The basic principle of this measurement system is that the output frequency of a tunable diode laser is adjusted so that as the measurement arm of a Michelson interferometer is scanned, there is no movement of the fringe signal². Figure 1 is a schematic of the optical apparatus and measurement and control strategy. The fringe signal arises in this apparatus from a differential measurement of the reference arm signal against the measurement arm signal. To track the laser diode frequency, a portion of the beam is split off and beat against a reference stabilized HeNe laser. This allows the diode laser wavelength to have a direct, unbroken traceability path. The beat frequency signal is logged synchronously with the acquisition of the STM images. This is a homodyne interferometer system and only a single frequency is used in the interferometer portion of the system. The two-frequency portion of the measurement is completely independent of the interferometer and measurement mirrors. In addition no fringe interpolation is required since the measurement is based on the number of fringes in the optical path difference between the reference arm and the measurement arm being held constant.

The calibration of the system is obtained by counting the number of fringes in the optical path difference between the reference and measurement arms. The number of fringes is determined by the wavelength of the diode laser and the path difference. This distance determines the scan range for the interferometer, when no mode hopping is allowed, based on the tunable diode laser range^{3,4}. Although mode hopping can be allowed, it is simplest to explain the principle with a fixed fringe measurement. By scanning the diode laser frequency, with a stationary measurement stage, and monitoring its wavelength based on the HeNe reference, the optical path difference can be obtained. This is accomplished by "mapping" a fringe of the differential intensity signal of the interferometer at the bicell detector while scanning the diode laser wavelength. The diode laser frequency is simultaneously beat against the HeNe laser. The bicell intensity signal of one full or half fringe is then plotted against the beat frequency signal. The map of the diode laser wavelength against the bicell detector intensity pattern results in a unique number of wavelengths making up the optical path difference. We discuss the hardware in more detail next.

Figure 2 shows a schematic of the interferometer and its equivalent Michelson interferometer configuration. The measurement mirror is scanned and the reference mirror is fixed to the isolated microscope stage. The beam is polarized at a 45° angle to the vertical. The beam passes through a non-polarizing beam splitter and travels into a birefringent crystal whose optical axes are at 0° and 90° to vertical. The beam is split as one polarization is bent towards the silvered reference mirror and the other travels straight through the crystal to the measurement mirror. The two beams bounce off of the reference

and measurement mirrored surfaces and reverse their direction. They travel back through the birefringent crystal and merge to become an elliptical beam at the birefringent crystal surface on exit of the crystal. They are then reflected at 90° in the non-polarizing beam splitter toward another birefringent crystal, which is at a 45° angle to the first birefringent crystal. This crystal is at 45° to mix the two polarization states, which formed the elliptical beam. This incoming beam is split by the second birefringent crystal into two different combined polarization states, each of which lands on the bicell detector. This differential method of measuring the beam intensity improves the signal to noise ratio and allows removal of any DC offsets. This output signal is logged and monitored for use in the control loop for the diode laser output frequency.

The diode laser frequency is continuously controlled to lock to a null point in the fringe signal measured at the bicell detector. As the measurement mirror is scanned, the diode laser frequency is changed to maintain a constant zero in the difference signal between the two polarization states exiting the 45° rotated birefringent crystal. The diode laser is operated with a control frequency of a few hundred hertz. The fundamental limitation is set by the piezo, which controls the frequency of the diode laser, since it has a maximum operation speed of 2 kHz. However, the limiting component in the current configuration is that the time interval analyzer (TIA) used in the frequency measurement cannot be read faster than 800 Hz. For most measurement applications, and with reasonably linear scanning velocities, these values are adequate and in fact go well beyond what our current applications require.

The optical path for the measurement of the diode laser frequency is also shown in Figure 1. A stabilized HeNe laser is used as the reference signal⁵. The tunable diode laser beam is split with a polarizing beam splitter. Half the signal goes to the reference measurement system and the other half goes to the interferometer. The HeNe reference beam and the diode laser beam come together in the reference measurement paths with orthogonal polarization states. The two beams are then mixed so that they interfere and the beat frequency can be monitored. The combined beam goes through lenses and a collimator and is fed to a microwave amplifier and into the TIA. This nominally 2 GHz signal then gives a direct measure of the output diode laser frequency relative to a stabilized HeNe reference signal, which has been appropriately calibrated. The output signal from the TIA is logged by computer in a burst mode with an appropriate trigger to synchronize the TIA data with the STM z height data.

We will now develop the equations to quantitatively evaluate the system performance and optical path difference mapping capability. The tunable diode laser, with a wavelength of nominally 632 nm, is introduced through a polarization-maintaining optical fiber. A portion of the original diode laser power of 3 mW is split off and used for monitoring the beat frequency of the diode laser against reference HeNe laser. As described above, when the optical path difference changes during a measurement, the frequency of the tunable diode laser is continuously controlled so that the number of wavelengths in the varying optical path difference is kept constant. Under this condition with the tuned laser wavelength λ , there is an integer number N of half wavelengths along the optical path difference (or the arm-length difference) ℓ , such that

$$N \frac{\lambda}{2} = n\ell \quad (1)$$

where n is the refractive index of the air along the optical path difference. For our system when operating in UHV, n is equal to 1.

Under servo control, the fringe signal is locked to a null point (zero crossing). The expansion $\Delta\ell$ of the optical path difference caused by a sample-scan along the x-axis results in a change in wavelength as follows:

$$N = 2\ell / \lambda = 2(\ell + \Delta\ell) / (\lambda + \Delta\lambda) \quad (2)$$

Equivalently,

$$\Delta\ell = (\Delta f / f) \ell \quad (3)$$

and ℓ can be obtained from an independent measurement of the frequency difference between different interference orders under a fixed optical path difference. The laser wavelength is swept while the optical path difference increases by an integer number m of half wavelengths as follows:

$$\ell = N \frac{\lambda_N}{2} = \frac{Nc}{2f_N} = \frac{(N+m)c}{2(f_N + \Delta f_m)} \quad (4)$$

Equivalently,

$$\ell = \frac{mc}{2\Delta f_m} \quad (5)$$

Equation 5 enables the direct mapping or determination of the optical dead path, ℓ , based on measuring the change in frequency required to yield one half cycle of the wavelength. Then, a displacement $\Delta\ell$ is obtained by monitoring the beat frequency between the diode laser and a reference laser. With this arrangement, an increase of 1 GHz in frequency is equivalent to a measurement mirror motion of 32 nm relative to the interferometer birefringent crystal (reference mirror) with a nominal value for ℓ of 14.5 mm. Atomic resolution on the order of 10 pm can be achieved because of the ability to perform highly accurate frequency measurements. The measurable range, 165 nm in this configuration, is roughly estimated by Eq. (3) with the optical path difference (14.5 mm) and frequency range of the counter (5.4 GHz, twofold range for a beat-frequency).

Polarization Mixing

In general, polarization mixing is a source of uncertainty in a fringe interpolation scheme of heterodyne interferometry⁶. The major causes of polarization mixing are the finite extinction ratio of a polarizing beamsplitter and an adjustment imperfection of the polarization axis to the beamsplitter. For the current homodyne Michelson scheme, we are not using heterodyne interferometry and hence there is no polarization mixing error. It is possible, however, that stray light from multiple reflections may disturb the fringe at the detector. The influence of stray light on the current frequency-tracking scheme is under investigation as well as polarization effects due to temperature changes in the birefringent crystals.

Conclusion

A new lightweight interferometer design and tunable diode laser measurement scheme was successfully implemented. The gains over conventional interferometry were outlined and the overall key components in the uncertainty were calculated to be 20 nm. We have preliminarily examined most of the uncertainty components in this system. A more detailed discussion is available⁷.

Although the interferometry methodology and system presented here was designed to measure distances such as atomic spacings or linewidths in the 100 nm range, the method can be expanded to measure larger distances. The key in this type of application is to allow mode hopping in the differential measurement of the interferometer bicell detector output. In this case, as the sample is scanned and the diode laser reaches its maximum or a pre-set value, the diode laser frequency is quickly swept back by a defined number of wavelengths. The scanning does not need to stop during this mode hop if the scanning is well behaved and linear. We have examined this method preliminarily and are able to move the diode laser piezo stack at near to 1 kHz, which is the expected limiting element. Consequently, if one allows mode hopping, and keeps track of the number of modes hopped and the change in the optical path difference, significantly larger scanning distances are possible.

Acknowledgements

We would like to acknowledge the Office of Microelectronics Programs, NIST for partial funding of this project. We would like to acknowledge Lowell Howard who worked on some of the fundamentals to this laser interferometer. Also, we are grateful for useful, in-depth discussions with John Kramar and would like to acknowledge J. Fu, H. Soons and E. C. Teague for helpful inputs.

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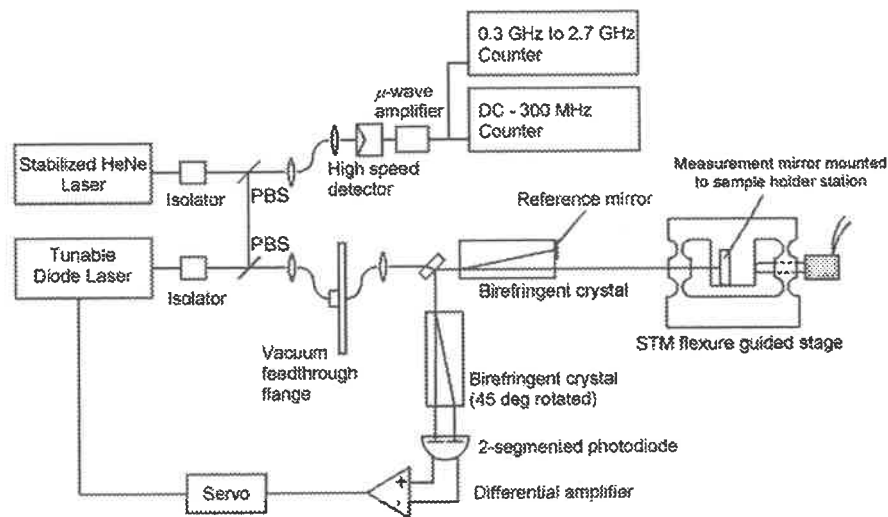


Figure 2. This shows the general methodology for tracking the measurement mirror motion and the general hardware functions. The reference measurement process is also shown between the HeNe and diode laser.

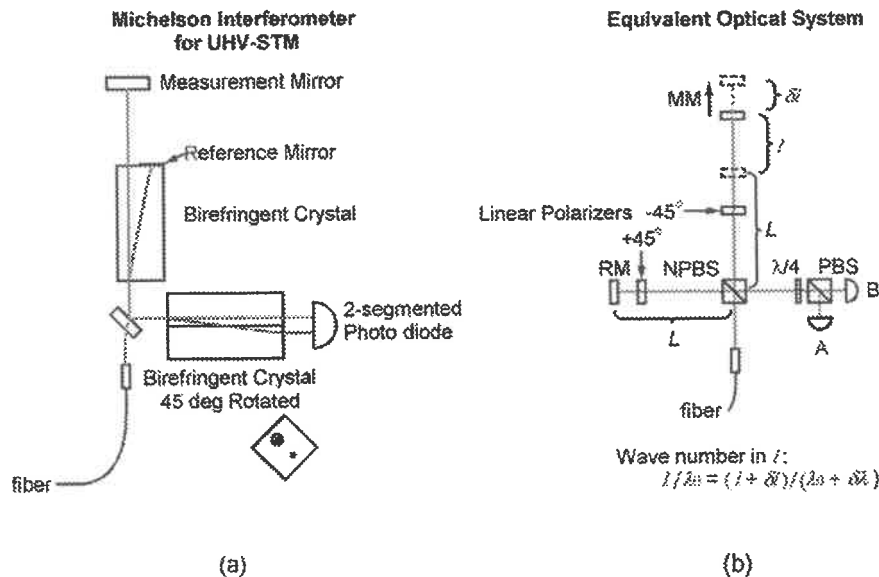


Figure 4. This is the interferometer configuration and the equivalent functional Michelson interferometer.

New Heterodyne Interferometric Method of Displacement Measure and Control with a Sub-nanometric Precision Over a Range of Several Millimeters

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Abstract: We present a new type of interferometric position control method with sub-nanometric accuracy. Our method is based on the use of a heterodyne Michelson interferometer and a high frequency clock. With this system, one can control the displacement of target mirror by a quantified step of value $\lambda/128$. As the relative uncertainty on the laser wavelength in vacuum is 1.6×10^{-9} (for $\lambda = 632.991\,528\,\text{nm}$), each step corresponds to a displacement of $4.495\,246\,313(8)\,\text{nm}$. This system is dedicated to the watt balance project of the national institute of metrology of France, the BNM. The aim of this project is to give a new definition to the mass unit by linking it to an invariant quantity – the Planck's constant.

Keywords: position control, laser interferometry, watt balance, metrology

1. Introduction

In the international system of unit SI, the kilogram is the only remaining base unit which still relies on a material artifact. This definition is unsatisfactory, especially because not much is known about the stability of the kilogram prototype. A new definition of the unit of mass is needed which is based either directly or indirectly on an invariant atomic process or fundamental constants. Various experiments are pursued worldwide which, in a first step, allow the monitoring of the kilogram and, later on, could serve as a basis for its replacement. At present, one of the most promising approaches is an experiment called the watt balance in which mechanical and electrical power are equated [1]. If the electrical power is measured in terms of the two quantum effect, Josephson effect and quantum Hall effect, the unit of mass can be linked to the Planck constant. A new type of watt balance is currently developed at the Bureau National de Métrologie (BNM, France) with the collaboration of several laboratories. The comparison between both powers needs two steps. In the first step, the weight of a mass unit m is counterbalanced by a Laplace force F_z induced by a current i passing through a coil of length l :

$$F_z = Bil = mg \quad (1)$$

To obtain a relative accuracy of 10^{-9} on the mass measurement, it is necessary to have the same level of accuracy for each physical quantity represented on Eq. (1). Now, this is not possible to measure the product Bl with such accuracy. That's why a second stage is needed. In this second step, the coil is moving with a velocity v and an electromotive force appears inducing a voltage ϵ between each end of the coil.

$$mgv = \epsilon i \quad (2)$$

The current i can be measured by comparison to a voltage standard based on Josephson effect and to a resistance standard based on quantum Hall effect.

$$mgv = \frac{A}{K_J^2 R_K} \quad A = \frac{f_1 f_2 i}{k} \quad (3)$$

where K_J and R_K are respectively the Josephson and the von Klitzing constants. Eq. (3) leads to a relation between the mass unit m and the Planck constant h :

$$m = h \frac{A}{4gv} \quad (4)$$

Hence, to measure m we need to measure two frequencies f_1 and f_2 , the acceleration of earth gravity g and the speed v of the moving coil. This speed is planned to be equal to 2 mm/s with an accuracy of 2 pm/s. In this paper, we present the method used to achieve such accuracy on the speed measurement. It combines the use of a heterodyne Michelson interferometer and a high frequency electronic circuit that allows us to lock in the optical phase at the output of the interferometer (*i.e.* the position of a mobile mirror) on the electronic phase of the high frequency circuit.

2. Description of the Phase-Locked Interferometric Method

Neglecting refractive index of air fluctuations, a displacement of a movable mirror given by a one-pass Michelson's interferometer by an amount of $\lambda/2$ corresponds to a phase shift equal to 2π . The method we proposed here consists in reversing this property of Michelson's interferometers. Consider the sketch on Fig. 1. An electronic board generates two synchronized signals s_1 and s_2 at the same frequency f . Electronic circuit allows us to make phase-jumps of quantified value on either signals. Signals s_1 and s_2 are respectively sent to a mixer and to a laser head which transpose the signal from ultrasonic range to optical frequency range thanks to a Bragg cell. This allows to perform the two optical components of the heterodyne interferometer. The optical beam passes through the interferometer. The two components are recombined at the output of the interferometer, resulting on a signal s_3 at a same frequency that s_1 and s_2 . Signal s_3 contains position information. Then s_3 is phase-compared with s_2 and a signal error is sent to a lock in electronic that pilot an actuator supporting the movable mirror.

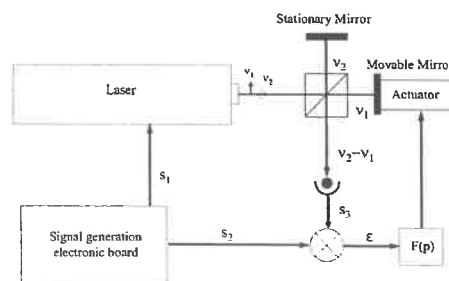


Fig. 1: Principle

3. Experimental Setup

Experimental setup is detailed on Fig. 2. The laser head allows us synchronizing the reference frequency f to an external 20 MHz signal through a fibered input. The internal 20 MHz signal is hence replaced by a 20 MHz frequency signal coming from our electronic board. Phase jump of $2\pi/32$ is induced by a home-made specific electronic circuit. This corresponds to a step of $\lambda/128$. The uncertainty on the step position value is directly related to the uncertainty onto the laser wavelength. The wavelength in air of the laser is given by $\lambda = c/nf$ where f is calibrated by beat frequency technique with regard to a national reference. The wavelength of the laser has been calibrated by Zygo and the nominal value is equal to $\lambda = 632.991\,528(1)\text{ nm}$ [2]. Hence we expect a step value of $4.945\,246\,313(8)\text{ nm}$. This value is the limit on accuracy that can be achieved with our method. One part of the beam issue from the interferometer is used to lock-in a piezoelectric actuator supporting the

movable mirror while other part is sent into electronic board of the commercial system to measure the displacement. Simultaneously, a weather station measures the room temperature ($\sigma_T = 5$ mK), pressure ($\sigma_P = 3$ Pa), humidity content ($\sigma_H = 1\%$) and CO₂ content ($\sigma_{CO_2} = 10$ ppm). These values allow us to measure the fluctuations of the refractive index of air using Edlén equations [3].

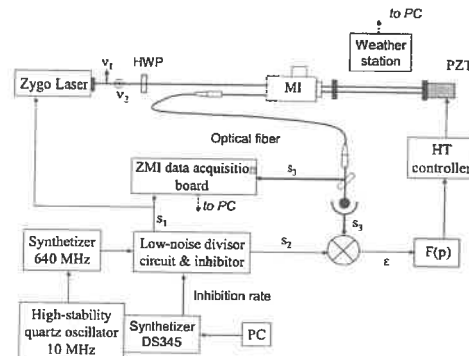


Fig. 2: Experimental Setup

4. Experimental Results

Step by step displacement of the target mirror is depicted on Fig. 3. We can see steps corresponding to each phase jumps at the mixer. Noise level on each step is equal to 0.22 nm (at 1σ). Fluctuations of the room temperature, pressure, humidity content and CO₂ content are respectively, 0.1°C, 100 Pa, <1% and 60 ppm. Using Edlén equations, refractive index of air variation is 17×10^{-8} . This corresponds to a negligible correction factor. Uncertainty on the speed of the movable mirror is inversely proportional to the number of steps. The displacement range of the PZT actuator used in our experiment is limited to 3 μ m for an applied voltage of 100 V. Note that our method do not have displacement range limit. This displacement is represented on Fig. 4. The slope of this curve is equal to the speed of the movable mirror. This speed θ is imposed by the period of repetition of the phase jumps equal to 20 Hz. As each step is equal to 4.945 nm, θ is equal to 98.904(1) nm/s. Uncertainty on θ is calculated with a step number η equal to 565 and a noise level on each step σ_p equal to 0.22 nm at 1σ . Relative uncertainty on the speed already reaches 1×10^{-3} . To reach a relative uncertainty of 10^{-9} , we need to make 3×10^5 steps that is to say 1.5 mm of displacement. Displacement range in watt balance project is planned to be 60 mm, so such uncertainty could be achieved with our method.

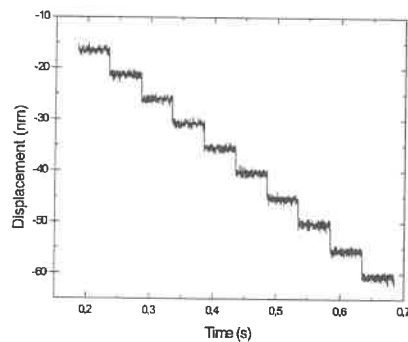


Fig. 3: Displacement measurement

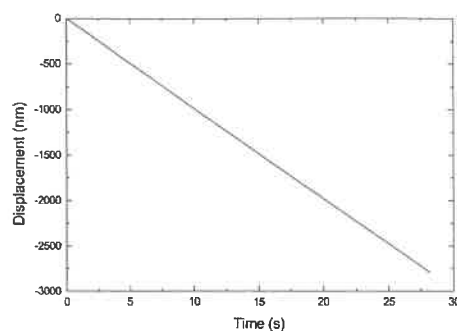


Fig. 4: Speed measurement

5. Conclusion

We have proposed a novel simple method to control step by step the displacement of a translation stage. Our method is based on the use of a heterodyne laser interferometer and a high frequency phase-jump electronic circuit. It allows to control the displacement of a target mirror with nanometric steps and sub-nanometric accuracy. This system of nanodisplacement is dedicated to the watt balance project of the Bureau National de Metrologie (France).

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DYNAMIC MATERIAL PROPERTIES FOR MACHINING SIMULATION USING THE NIST PULSE-HEATED KOLSKY BAR

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Abstract

The modeling or simulation of machining processes is important for economically implementing new manufacturing processes such as high-speed milling or ultraprecision machining. Knowing if a process will provide the required output prior to investing in expensive equipment is vital. All machining process models, including those for ultraprecision processes, are dependent on knowing the appropriate material properties. The NIST pulse-heated Kolsky bar is an experimental apparatus built for measuring the dynamic material properties at elevated temperatures useful in machining models. Stress-strain curves for AISI 1045 steel obtained at strain rates of over 2000 /s and high temperature (over 900 C) are presented.

1. Introduction

The need to improve machining processes has been obvious to machine shop workers for hundreds of years. Great strides in optimizing processes have been accomplished by many generations of machinists who boldly varied parameters and implemented improvements based on their direct experiences. Early manufacturing researchers realized that it would be extremely useful if one could calculate important parameters, such as cutting forces and tool temperatures, by plugging into a formula. However, it was recognized that material removal processes are extremely complicated and that scientific research was necessary. Some of the early attempts at understanding machining from a scientific standpoint were Ernst [1] and Merchant [2] in the first half of the twentieth century. Then by the 1970s, hundreds of research papers had been published. Several different historical surveys, for example Komanduri [3], summarize much of this work. A variety of detailed methods were developed and presented, but by the late part of the century it appeared that the most promising analytical method of predicting machining process parameters was the finite element modeling (FEM) method. Some of the earliest work of applying the FEM method to machining research came from the precision engineering community, with work by Strenkowski at North Carolina State in the 1980s [4].

The FEM method can be used to examine the effects of different tool geometries and parameters, such as cutting speeds and feeds. Detailed and user-friendly codes have been developed; for example, work presented by Marusich [5] describes an approach that is used in a commercial code.* Temperature profiles can be calculated as part of the FEM approach for even the small depths of cuts used in the ultraprecision machining processes, such as diamond turning, with virtual spatial resolution well beyond what can be physically measured. Even though it is easy to scale the geometry to the small depths of cuts by the FEM approach, there are still many unanswered questions concerning the capability of the models to handle the physics of small depths of cut. Research into the size effects for machining process modeling will continue and this future work will be very important to the precision engineering community. FEM modeling will undoubtedly contribute to the advancement of ultraprecision machining in the future as it is currently contributing to the improvement of more conventional machining processes.

* NIST is a US Federal Laboratory and does not endorse any particular manufacturer's product. Examples are given only to illustrate the types of products available.

An essential part of FEM methods is a material constitutive model that relates stress to strain. FEM codes have the advantage of being able to handle complicated representations of the constitutive model that include the high strain, high-strain rates, high temperatures, and high-heating rates found in machining processes. An FEM analysis with many elements can involve a very large number of relatively simple calculations. With modern fast computers, realistic geometries and complicated constitutive relationships can be handled in reasonable times. Practical difficulties arise in applying the FEM approach because the necessary constitutive models are not readily available for materials often required in modern manufacturing. Also, the accuracy of the FEM results are often limited by not being able to accurately describe the constitutive behavior of the material under the conditions found in real machining processes.

2. The NIST pulse-heated Kolsky bar apparatus

To assist in ongoing efforts to provide constitutive model data needed for FEM codes to adequately simulate machining processes, we have built a new facility at the National Institute of Standards and Technology (NIST). This facility is based on a well-established method for obtaining high-strain-rate stress-strain curves called a Kolsky bar (or split Hopkinson pressure bar, see the ASM Handbook articles by Follansbee [6] or Gray [7].) To this traditional apparatus we have added a unique capability of resistively heating the sample immediately prior to high strain rate testing. The resistive heating is produced by an accurately controlled pulse of DC electric energy delivered to the test sample from a special apparatus developed by the NIST Subsecond Thermophysics Laboratory. The pulse heating system and Kolsky bar apparatus are described in Yoon et al [8] and Basak et al [9], and the construction and alignment of the NIST Kolsky bar was presented at the 2002 ASPE Annual meeting by Rhorer et al [10]. The Kolsky bar can provide compression testing at strain rates from about 500 /s to 10 000 /s

(traditional tension or compression tests can be done at rates from .001 /s to approximately 100 /s).

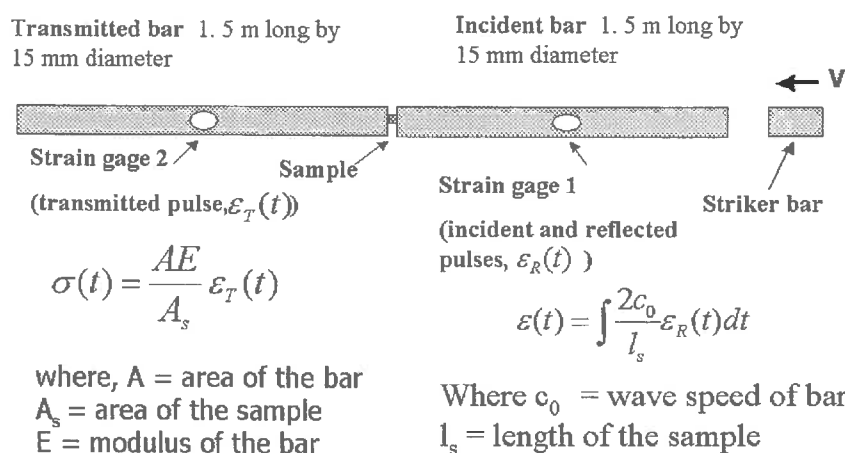


Figure 1. Kolsky bar operation

The operation of the traditional Kolsky bar is outlined in Figure 1. By recording the output of the two strain gages the stress-strain curve for the sample material can be easily determined. The 15 mm diameter by 1.5 m long bars are supported by guide bearings to support and maintain the alignment of

the bars as in the traditional Kolsky apparatus. In the NIST system the support bearings are all plastic except the center two bearings, which are graphite lined metal bearings. The center two bearings are connected to a DC electric power supply through large welding cables. The 4 mm diameter by 2 mm long sample, sandwiched between the two long bars, is then resistively heated as the current passes through short sections of the bars and directly through the sample. The volume of the sample is less than 0.01 % of the volume of the portion of the bars conducting the electricity to the sample, and therefore the sample heats much more rapidly than the supporting bars.

A test consists of programming a temperature heating pulse (for example set to reach 1100 K) with a heating time from 150 ms to several seconds. The air gun is triggered with an adjustable delay circuit so that the impact test occurs at the end of the heating pulse. In a typical test, the strain wave generated by

the impact of the striker bar compresses the sample in approximately 150 μ s. During the test the sample temperature is monitored by a high-speed pyrometer (with one μ s resolution) and the uniformity of the sample heating is verified by a thermal imaging camera (approximately 3000 frames/s). The strain gage output is recorded on a digital oscilloscope using standard strain gage circuitry calibrated by the parallel resistor method.

3. Results and Discussion

The resulting stress-strain curves from the Kolsky bar tests can be generated using the approach outlined in Figure 1 and the strain gage recordings. Some results of tests on AISI 1045 steel are shown in Figure 2. These curves show the true stress (flow stress) as a function of plastic strain. At lower strains (less than approximately 3 %) the plots are not accurate because the Kolsky bar technique is not a useable test for determining the elastic modulus of the material but determines plastic strains up to 20 % or more. The upper three curves are from tests at room temperature (without a heating pulse) and illustrate the effect of strain rate on the flow stress. In general, the higher the strain rate, the higher the flow stress.

The flow stress at elevated temperatures is lower (thermal softening) as shown by the two lower curves in Figure 2. For example, the curve labeled Test 305 was a test where the sample was heated to 620 C in approximately 700 ms (a heating rate of approximately 900 K/s) and then held at the temperature for 50 ms prior to the impact. Ongoing and future work will look at additional temperatures and heating rates as well as the repeatability of the tests for the steel samples. In addition future work will include expanding

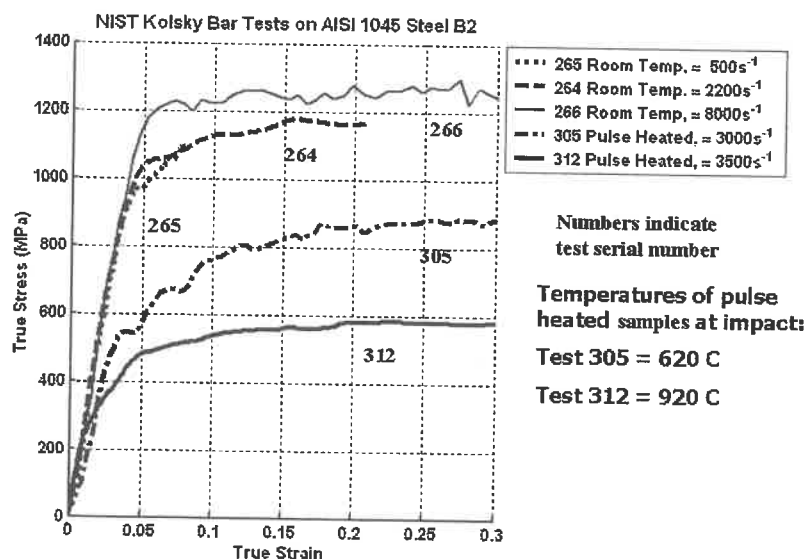


Figure 2. Results of Pulse-heated Kolsky bar tests.

is fast enough, microstructure changes will not have time to occur prior to the large deformation caused by the impact and rapid compression of the sample. Therefore, in the case of rapid heating the thermal softening should be less than the case of heating to the same temperature but at a slow heating rate. Preliminary high-strain-rate (~ 4000 /s) stress-strain data for AISI 1045 steel samples heated to above 900 C in 250 ms (a heating rate of ~ 3600 K/s) indicate a higher flow stress (~ 200 MPa higher) than samples heated to the same temperature in approximately 2.5 s (~ 360 K/s rate). Additional tests are being performed to further examine and verify the heating rate effect.

the testing efforts to other materials of interest in machining simulation. Standardizing some of the methods for testing and data reporting of the dynamic material properties are also being investigated as part of this project.

One of the questions that can be addressed with the NIST pulse-heated Kolsky bar, which is not possible with more conventional furnace type heating of the sample, is the effect of heating rate on the stress-strain curves. If the heating rate

4. Conclusions

Process modeling has become an essential component in modern manufacturing enterprises. To develop reasonable models for the machining processes it has been necessary to develop a means of measuring dynamic properties of the materials being machined. Because machining processes involve both high strain rates and high heating rates, the NIST pulse-heated Kolsky bar has been developed as a system to measure these material properties and to provide a test bed for materials research. Using the new pulse heated Kolsky bar apparatus we have demonstrated that stress-strain curves for strain rates from 500 /s to 8000 /s can be generated with a sample temperature of over 1100 K and heating rates of over 5000 K/s immediately prior to the mechanical tests. Work is continuing to expand these ranges of strain rates and heating rates to more completely cover the range of parameters found in machining.

5. Acknowledgements

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Direct Single Point Diamond Cutting of Stavax Assisted with Ultrasonic Vibration to Produce Optical Quality Surface Finish

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1. Introduction

Steel is widely used in the plastic molding industry. However, it is well known that steel cannot be machined using diamond tools due to the excessive tool wear, which is mainly caused by diamond graphitization [1-4]. In vacuum, diamond is converted to graphite when the environmental temperature is over 1800°K. If iron elements are present, the conversion temperature is reduced to between 400°K and 600°K. To minimize diamond graphitization when machining steel, various approaches have been attempted. High carbon steels were cut in a carbon rich environment (methane and ethyne) [4, 5]. It was found that a carbon rich environment helped to reduce the diamond tool wear rate. Evans [6] observed that the tool wear was much reduced when the cutting area was cooled using cryogenic method. Ultrasonic vibration assisted machining has been used for cutting, grinding, and tapping by Kumabe since 1960s. He suggested that steel could be machined using diamond tool by applying ultrasonic vibration [7]. However, it was only recently that ultrasonic vibration assisted machining has been successfully used in ultra-precision machining. Moriwaki and Shamoto [8] developed an ultrasonic vibration assisted cutting (USC) system to cut steel to mirror surface finish. The tool life was also increased significantly. Klocke and Rubenach [9] built a similar device and used it to make steel parts for industrial applications.

This paper reports on the use of a USC system for theoretical and experimental studies of the lubrication of the tool-chip interface.

2. Experimental Set-Up

The experiments were conducted on a Precitech ultra-precision lathe (Optimum 4200). A mixture of compressed air and baby oil was used as the lubricant. The work material was Stavax with a hardness of HRC 40 to 45 (stainless steel, AISI 420, 0.38% C, 0.9% Si, 0.5 Mn, 13.6% Cr, 0.3% V). It is widely used as molding tools on account of its high strength, corrosion resistance and machinability.

An ultrasonic vibration system was constructed as shown in Fig. 1. The ultrasonic vibration unit was mounted on a height-adjusting frame with a clamping ring. One clamp position is at the static node of the vibration wave. To compensate for the poor rigidity with only one static clamp point, a dynamic clamp point was introduced at the free end of the horn. This will strengthen the rigidity in the thrust direction. The arrangement would largely eliminate the lateral vibration. The effect on the longitude vibration is minimal. The diamond tool was mounted at the end of horn and vibrated in the direction of the horn axis -the cutting direction- with a frequency of 40 kHz. The vibration amplitude was adjustable from 2 to 24 μm .

A precision non-contact capacity ADE Microsense gauging system was used to measure the amplitude of the ultrasonic vibration. The vibration signals were analyzed by Fast Fourier Transform (FFT). The machined surface roughness was measured using a Wyko white light interferometer.

3. Lubrication during Ultrasonic Vibration Assisted Cutting

For each vibration amplitude, the cutting speed has to be properly selected to achieve mirror surface finish. Researchers have suggested that the cutting speed need to be only below the maximum vibration speed. Our investigation showed that the practical cutting speed to achieve mirror surface finish

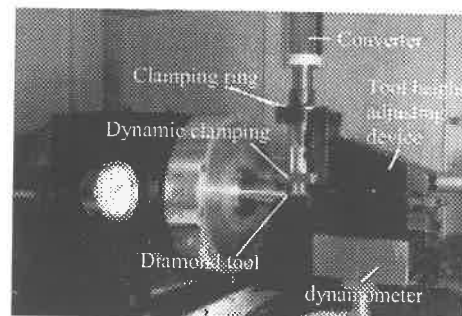


Fig.1 Ultrasonic vibration assisted cutting system

can be significantly lower than the maximum vibration speed. To understand this, an analysis of the tool-chip gap and the lubrication at the cutting zone was undertaken.

During ultrasonic vibration assisted cutting, the workpiece moves towards the cutting tool, with the cutting tool vibrating at a frequency f and amplitude A , as shown in Fig. 2. When the diamond tool is in the retraction part of the vibration cycle, the vibrating speed increases rapidly. Once the speed is higher than the cutting speed, v , the tool rake face starts to separate from the chip, say, at point a . The gap between the tool and chip, Δd , increases as the tool retracts further. Once the tool has passed the neutral point, the vibration speed slows down. When it is again equal to the cutting speed on the other half of the cycle, the gap reaches its largest value at point c . As the vibration speed slows down, the gap, Δd , decreases from the maximum value to zero at point b . The tool vibration motion can be expressed with the formula:

$$y = A \sin \omega t \quad (1)$$

where $\omega = 2\pi f$, and t is time.

The gap between the tool and chip is given by:

$$\Delta d = A(\sin \omega t_a - \sin \omega t) - v(t - t_a) \quad t_a < t < t_b \quad (2)$$

where t_a and t_b are t at point a and b , respectively.

After the differentiation of equation (2) with respect to t , and simplifying, the times for equal speeds at a and b is:

$$t = \frac{1}{\omega} \cos^{-1} \left(\frac{v}{2A\pi f} \right) \quad (3)$$

The maximum gap can be calculated with equation (2) once t_a and t_b are found. Fig. 3 shows the variation of the maximum gaps between the tool and the chip with the cutting speed at different vibration amplitudes. The maximum gap is found to decrease almost linearly with an increase in the cutting speed. When the cutting speed is very low, the maximum gap is around twice the value of the vibration amplitude. When the cutting speed reaches the maximum vibration speed, no gap exists between the tool and the chip.

To simplify the analysis of the lubrication mechanism at the tool-chip interface, some assumptions can be made. As the tool nose radius (1mm) is much larger than the depth of cut (6 μm), the tool contact arc is considered to be a straight line. Hence, $OABC$ is the contact area between the tool and the chip (see Fig. 4). OA is the tool-chip contact length. It is further assumed that the coolant fluid enters the gap only from the side AB , and no coolant enters from CB or OA , since AB is much longer than CB and OA . There is also no coolant coming from OC , which is the cutting edge.

When the rake face retracts from the chip by dz , the coolant advances by $dl \sin \kappa$ towards the cutting edge, in the negative y direction. The volume of the vacuum created should be equal to the volume filled in by the coolant; that is:

$$l_o a_c dz = l a_c dz + a_c h dl \quad (4)$$

$$(l_o - l) a_c v_s dt \sin \kappa = a_c h u_f dt \quad (5)$$

$$\frac{u_f}{\sin \kappa (\ln h)} = l_o - l \quad (6)$$

where κ is the cutting edge angle; l_o is the tool-chip contact

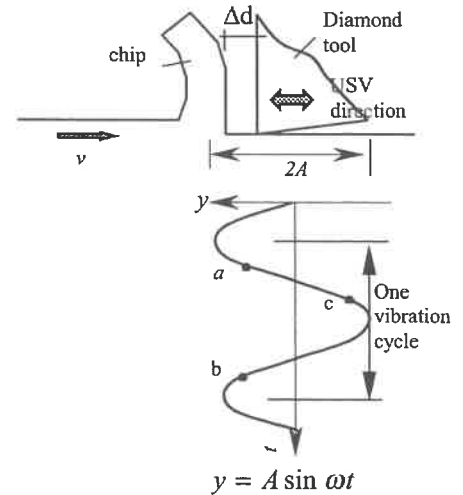


Fig. 2 Ultrasonic vibrating of diamond tool

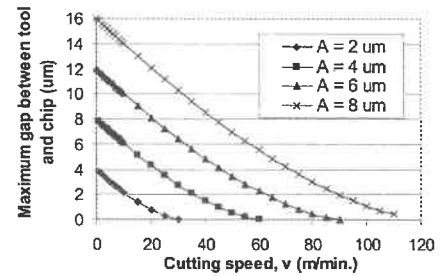


Fig. 3 Maximum gaps between tool-chip vs cutting speed

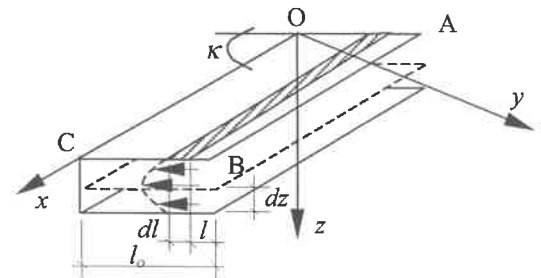


Fig. 4 Model for coolant filling into tool-chip

length; l is the distance advanced by the coolant at time t ; a_c is the depth of cut; h is the gap between the tool and the chip at time t ; v_s is the separation speed of the tool from the chip ($= h'$); and u_f is the average speed of the lubricant moving front at l .

To eliminate the variate l , differentiate equation (6) with respect to time t , and simplify it. Hence:

$$u_f = C_1 \frac{h'}{h^{1-\sin \kappa}} = C_1 \frac{-A\omega \sin \omega t - v}{(A(\sin \omega t_a - \sin \omega t) - v(t - t_a))^{1-\sin \kappa}} \quad (7)$$

To determine the value of the lubricant advancement over time t , the above equation is integrated:

$$l = Ch^{\sin \kappa} = C(A(\sin \omega t_a - \sin \omega t) - v(t - t_a))^{\sin \kappa} \quad (8)$$

where C is a constant. From equations (7) and (8), it is observed that in initial stage, the lubricant moves very fast and then slows down.

4. Results and Discussion

Theoretically, using equation (8) it is able to determine whether the lubricant fully fills the tool-chip gap and lubricates the tool rake face by comparing the calculated l and the tool-chip contact length, l_o . However, the presence of the constant C in equation (8) limits its direct application to predict the coolant filling condition. From experiments, it was found that when the cutting speed was 5 m/min and the vibration amplitude 2 μm , depth of cut 6 μm and feedrate of 10 $\mu\text{m}/\text{rev}$, a machined surface finish of 8 nm Ra can be achieved (see Fig. 5). This shows that the coolant has reached the cutting edge and fully lubricated the tool. The maximum gap computed using equation (2) is about 3 μm . This condition can be used as the criterion to study the coolant filling effect under the same conditions except for the cutting speed and the vibration amplitude. The following equation can be used for the comparison:

$$R_o = l/l_o = \left(\frac{(A(\sin \omega t_a - \sin \omega t) - v(t - t_a))^{\sin \kappa}}{3} \right) \quad (9)$$

where l_o is the tool-chip contact length for the cutting conditions mentioned above. If R_o is larger than 1, the corresponding cutting speed is considered as the critical value. Fig. 6 shows the calculated critical cutting speed at which the tool-chip gap is fully filled for a given vibration amplitude. It can be seen that the critical cutting speed increases almost linearly with an increase in the vibration amplitude.

In the experiments it was difficult to ascertain whether the tool-chip gap was fully filled by direct observation. Instead, the finish of chip underside was used to judge the effect of lubrication. If there were no tear marks on the machined surface (see Fig. 7(a)), the tool was considered to be well lubricated and cooled. Based on this, the experimental critical cutting speed was obtained and plotted in Fig. 5. It can be seen that when the vibration amplitude is below 5 μm , the calculated results are in good agreement with the experimental values. However, for high cutting speeds the differences can be quite large. This is because equation (9) only considers the filling condition of the coolant, but does not take into account the heat which it is being conducted away. In high cutting speed, the cutting temperature at the cutting region can be very high. If the coolant does not stay long enough between the tool and the chip and insufficient heat is conducted away, this results in work material adhering to the cutting edge and results in a poorer surface roughness. Furthermore, the high temperature accelerates the diamond graphitization and exacerbates the tool wear. The prevalence of material adhesion can be conveniently deduced by examining the underside of the chips. At low cutting speeds the cooling time was sufficient and the underside of the chips were found to be very smooth (see



Fig. 5 Machined steel parts with mirror finish

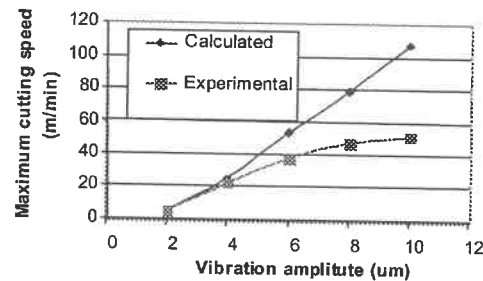


Fig. 6 Maximum cutting speed for a given vibration amplitude

Fig. 7(a)). This implies that there is little or no work material adhesion problem. On the other hand, grooves and tear marks could be clearly seen on the underside of the chips at high cutting speeds (see Fig. 7(b)). This suggested that material adhesion occurred at these cutting conditions. The lubricant was unable to reach to the cutting edge or, there was insufficient cooling time.

From the above analysis, it appears that increasing the amplitude while maintaining the same cutting speed can improve the lubricating and cooling effects. However, large vibration amplitude generates more heat at transducer connections and this can damage the vibration transducer through overheating. Therefore, low vibration amplitude ($<4\ \mu\text{m}$) and low cutting speeds ($< 20\ \text{m/min}$) are usually preferred.

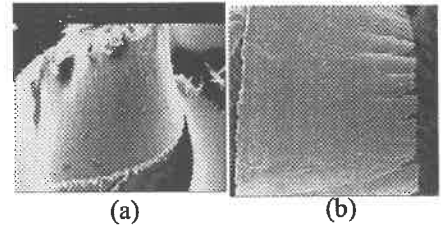


Fig. 7 chip underside with (a) sufficient cooling (b) insufficient cooling

5. Conclusions

From the above study, some conclusions can be drawn:

- An ultrasonic vibration assisted cutting system using a static and a dynamic clamping points provides an adequate stiffness in the lateral direction.
- The maximum gap between the tool and the chip decreases almost linearly with an increase in the cutting speed.
- The coolant filling speed is much faster at the early stage than the later stage of the vibration cycle.
- The calculated and experimental critical cutting speeds were found to be in good agreement at low vibration amplitudes. However, large differences existed at high vibration amplitudes due to the prevalence of work material adhesion on the cutting edge.

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CHARACTERIZING RESIDUAL STRESS IN SCRIBES ON SILICON USING DEFLECTION MEASUREMENTS Supplemented With Raman Characterization

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1.0 INTRODUCTION

It has been well documented that single crystal silicon undergoes phase transformation with the generation of high pressure on the surface by micro-indentation and diamond point scribing. Investigations of the transformation region created report phase transformation from diamond cubic (dc) structure to beta-tin phase. The transformation to the metallic beta-tin phase is thought to be responsible for anomalous plastic flow behavior without fracture seen in contact loading experiments. The beta-tin phase is not generally seen at room temperature or pressure but is rather identified by an amorphous phase or R8/BC8 crystal structure seen after unloading depending on experimental conditions. The implication of the ductile behavior is, that using careful machining conditions, single crystal silicon may be ground or diamond turned in the ductile regime. It is hoped that ductile machining will reduce cracks and dislocations generated by machining on the surface and subsurface regions that can reduce the mechanical integrity and lower the operation lifetime of the components.

A scribing apparatus was used to translate a diamond tip across a surface at a constant rate for different loads. With the appropriate conditions, the diamond tip is used to produce scribes on single crystal silicon samples generating brittle-to-ductile transformation of the material beneath the scribe. This anomalous plastic flow has been attributed to a high pressure phase transformation to a metallic β -tin phase.[1] The residual stresses within the scribe traces that result from the elastic-plastic constraint of the displaced material create a bending distortion in a flat-plate geometry wafer sample. The deflection is measured and extracted using optical interferometry techniques. The angle of deflection in the flat plate geometry correlates to the residual stresses created using a bend effect model developed for previous scribing research. Stress approximations will be coupled with Raman, SEM, and AFM characterization to aid in understanding the nature of the material deformation within the scribe zone.

2.0 PROJECT DETAILS

This investigation entails collecting bend data as well as Raman measurements for (100) and (111) crystallographic orientations of silicon, scribing with a Dynatex tip. Scribing along the $\langle 100 \rangle$ and $\langle 110 \rangle$ will be studied for the (100) orientation. $\langle 110 \rangle$ and $\langle 112 \rangle$ directions will be studied for the (111) orientation. The deflection angle data will then be correlated to the residual stresses created by scribing using a bend effect model. Indenting was also performed using Vickers and Dynatex tips to study the indentation regions for phase transformation with Raman spectroscopy.

2.1 EXPERIMENTAL SETUP

A Zwick micro hardness tester is fitted with a Dynatex diamond tip (Figure 1a) in a custom holder. A motorized stage is used to create translation at a constant rate of 0.250 mm per second. A holder was designed to hold the cutting edge of the diamond tip at an angle of 3.3 degrees from the sample surface as shown in Figure 1b. The

desired load is placed on the loading sleeve and the tip is positioned and contacted to the surface of the sample. The stage is translated with relation to the diamond tip producing the scribe. This setup can achieve repeatable loading below 50 mN and with relatively easy tip alignment in the translation direction.

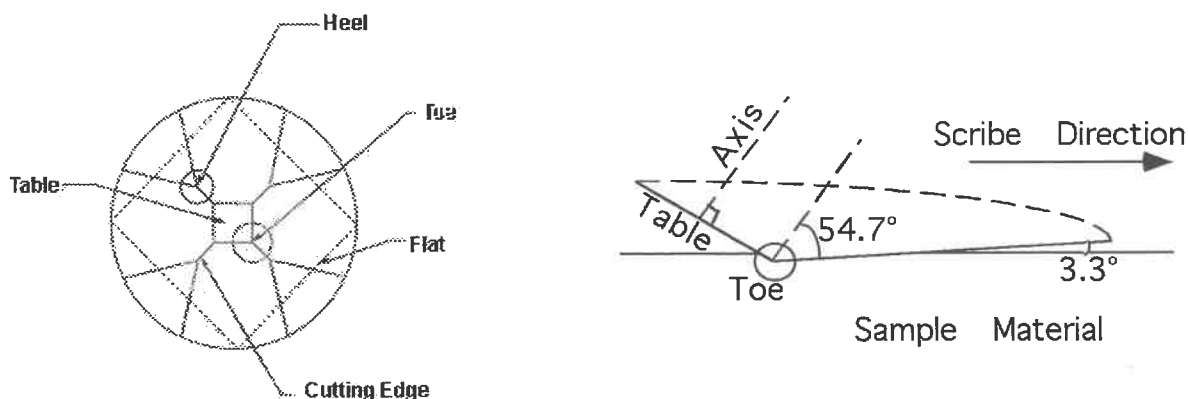


Figure 1. (a) Schematic of Dynatex Tip (b.) Geometry of Tip Relative to Surface [2]

Topographic data is collected for a 1cm x 2cm x 525 μm sample using a ZygoGPI Laser interferometer before and after scribing. Multiple, evenly spaced (50-100 μm) scribes are placed on the surface to generate measurable deflection (the effects of multiple scribes are additive to the bend) and to produce a more statistically significant measurement than that of a single scribe. The resulting profile created is the net bending effect, this is used to approximate the angle of deflection from the initial state.

Characterization of the scribe/indent region was accomplished with Micro-Raman spectroscopy done at room temperature using an ISA U-1000 scanning monochromator.

Raman excitation was done with the 514.5 nm line of an Argon-ion laser, with a spot size of approximately 3 to 4 μm in diameter. Raman spectra were taken in the 200-600 cm^{-1} range which contains the characteristic peaks normally associated with crystalline, amorphous, and various metastable crystalline phases seen in other research. [1]

2.2 EXPERIMENTAL RESULTS

Scribing with the Dynatex tip across a 2 cm x 1 cm x 525 mm plate of single crystal silicon yields a measurable deflection on the order of tenths of microns. This behavior is seen in of each of the crystallographic direction studied for both orientations if the load applied is enough to produce cutting action in the material.

Figure 3 shows two SEM images distinguishing between the two general responses. The image on the left is scribed on (100) silicon in the [011] direction at 5g load and appears to have no fracture around the scribe and cutting action appears plastic. The plastically displaced material, whether driven by the β -tin metallic phase or not, is pushed by the tip into the surrounding elastic material creating residual stress and deflection in the sample geometry. The image on the right was done at 30g load and shows a large amount of fracture. The action of the diamond indenter may create localized stresses at the tip which exceed the fracture limit, but it is unclear how the transformation region affects this. One of the limitations inherent is that with elastic fracture, removal of the plastically deformed regions may be possible and the bend effect model will not accurately give measure of residual stress for scribes. The bend effect will be reduced relative to a scribe with no fracture.

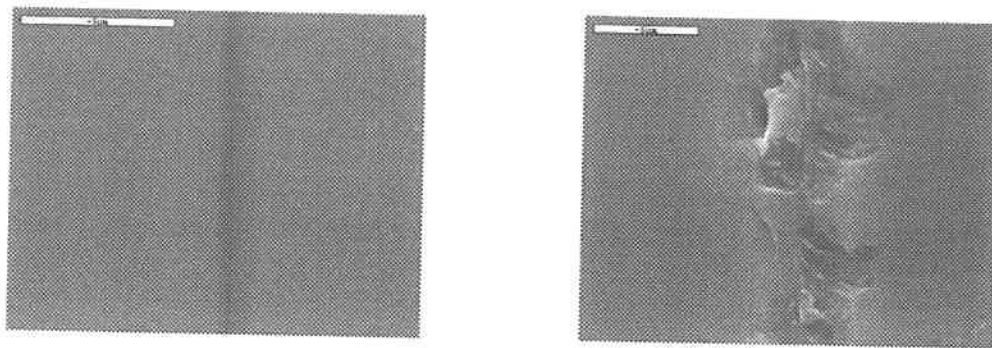


Figure 3. SEM images of scribes (100)[011] at 5g and 30g respectively.

Scribing between 1-20g in the various crystallographic directions with a sharpened tip shows an approximately linear relationship between the normal load applied to the tip and the deflection generated by the residual stresses. Figure 4 shows this trend for (100)[011] and (111)[-1-12], respectively.

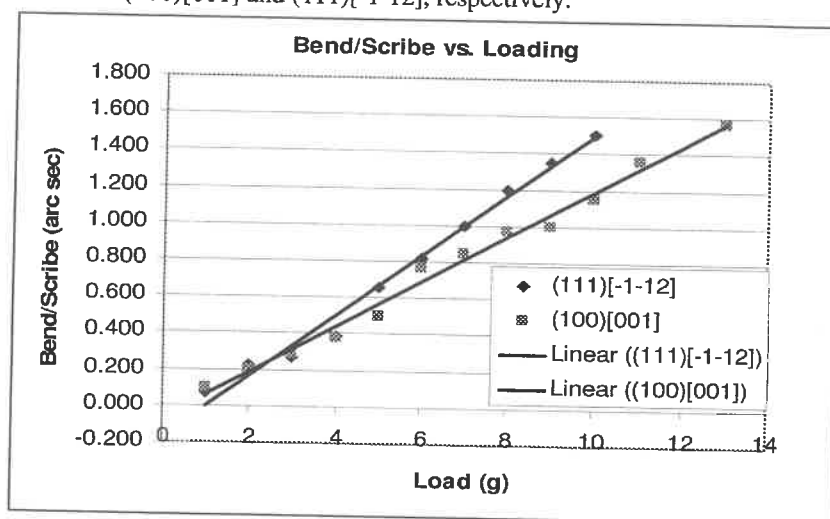


Figure 4. Bend angle generated per scribe versus load applied at tip.

The scribes for the data presented in Figure 4 appear to be fracture free along the scribe traces. These two directions represented the best cutting directions seen for their respective orientations. The linear trend could indicate a correlation between the load applied and the amount of plastic deformation occurring beneath the scribe. Experiments for repeatability of these results are currently underway. Other combinations of crystallographic direction and orientation are to be tested.

Another point of interest to note is the difference between results previously obtained during diamond point turning (DPT) of silicon wafers and those presented here from scribing with the Dynatex tip. Blackley et al. showed that for DPT, the [100] direction on (100) produced more favorable cutting behavior (fracture free, plastically generated) than the [011] direction [3.] For scribing with the Dynatex tip the opposite seems to be true. Figure 5 shows bend angle generated per scribe versus load applied at the tip. Optical microscopy revealed the onset of fracture for [011] to occur at 15 g with the Dynatex tip. The fracture threshold for the [100] direction on the other hand occurred around 7.5 g. The data in Figure 5 reflects the unpredictability of using deflection measurements for the [100] direction above 5 g in the prediction of stresses, while the trend seems to be linear up until the fracture point for the [011] direction. This is attributed to fracture that removes plastically deformed regions, thereby reducing the residual stresses and bend angles for the [100] direction.

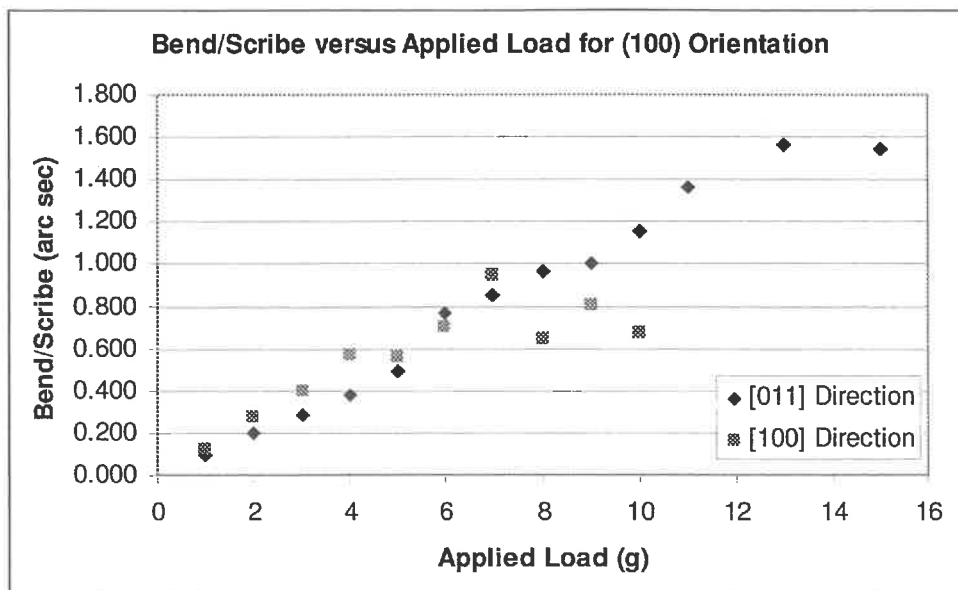


Figure 5. Bend angle generated per scribe versus load applied at tip.

Raman was performed on both Vickers scribes (1-15g) and Dynatex scribes (2-40g) with no apparent sign of peaks corresponding to the phases commonly seen after scratching tests. It is not known to the authors at this point why this has occurred. It could be possible that the phase transformation may not be occurring with the Dynatex tip due to the nature of the geometry tip and that the plastic displacement is by some other mechanism, for example, dislocations. It is also possible that the tip may be removing transformed material as it passes. The detection proficiency of the Raman unit may be responsible as well. If the penetration depth of the laser is much greater than that of the transformation zone, or if the laser spot size is large compared to the scribe width, the Raman spectra for the transformation product may not be detected. Sufficient data is not yet available to draw a conclusion for scribing experiments. Indenting experiments are currently underway to test the Raman unit in indenting conditions already known to generate phase transformation. [1]

3.0 CONCLUSIONS

The bend angle generated due to scribing is on order of arc sec per scribe depending on load applied to the tip. The deflection appears to increase linearly for scribes with minimal fracture, but hasn't shown to be repeatable for specific values of bend angle with similar applied loads. For worn tips bend angle drops dramatically (an order of magnitude) at certain point "threshold." Raman measurements yielded no sign of phase transformation within the scribe regions for both Dynatex and Vickers geometries, contrary to what was anticipated by the authors. This is still being investigated. Work continues with using dead-weight loading to create scribe traces with better experimental procedure that aim to improve current data and repeatability. The bend effect model used to correlate sample deflection to an estimate of residual stress within the scribe trace is being tailored to this investigation. Indenting experiments are currently underway to reproduce the experimental findings of others [1] and gain better insight into the lack of Raman data indicating the expected phase transformations within the scribe region. Under development is a piezo-actuator driven indenting apparatus. This will aid in performing controlled indents and load control during scribing to test the effects of loading/unloading rate of the tip into the surface.

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Effect of Machining Parameters on Machining Performance of Micro EDM and Surface Integrity

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ABSTRACT

Micro Electrical Discharge Machining (EDM) is one of the micromachining methods. It can be used to machine micro features and micro parts. This paper presents the details of an experimental study and related analysis of the effect of machining parameters on machining performance of Micro EDM under various conditions. Examples of 3D complex micro cavity, non-circular blind hole and micro hole with high aspect ratio machined by Micro EDM are presented.

INTRODUCTION

In Micro EDM, the material is melted and vaporized due to high temperature generated by an electrical discharge with small energy in the interelectrode gap immersed in a dielectric liquid. The molten material is ejected at the end of the pulse due to the collapse of the plasma column, resulting in a crater generated at the workpiece surface as well as on the tool surface. The machining parameters such as voltage, pulse duration and dielectric influence the process performance (including material removal rate, electrode wear ratio, gap and surface integrity). The debris concentration in the dielectric at the very narrow interelectrode gap often leads to the occurrence of abnormal discharges and short circuits, resulting in high electrode wear and low material removal rate.

This paper reports a summary of experimental results of the effect of voltage and capacitance on material removal rate, electrode wear ratio, discharge gap and surface roughness [1,2]. These experiments have been conducted on a commercial Micro Electrical Discharge Machining (EDM), using voltage values from 70 to 100V and capacitance of 10, 100, 220, 1000, 3300pF. In contouring EDM, slots were machined layer-by-layer. The layer depth was changed from 1 to 5micron to observe its influence on material removal rate and electrode wear ratio. The workpieces were of stainless steel and silicon. The experiments were conducted to generate micro-holes, -slot and -3D cavities [1,2].

EXPERIMENTAL RESULTS AND ANALYSIS

Effect on MRR

Material removal rate (MRR) is the removed volume of workpiece material divided by time. Generally, MRR mainly depends on the discharge pulse energy

(a function of capacitance and voltage) [3]. The larger pulse energy results in higher MRR. Figure 1 shows the increase of MRR with an increase of capacitance. The increase in voltage slightly increases the MRR as shown in Figure 2. Figures 3 and 4 show that the MRR can also be increased by increasing the tool moving speed, layer depth and capacitance in contouring Micro EDM. When drilling a very deep hole, the ejection of debris from the machining area becomes difficult. The debris concentration causes the abnormal discharges. One solution to this problem is the application of planetary movement to the electrode. It reduces the occurrence of abnormal discharges, and thus improves the MRR as shown in Figure 5.

The type of dielectric also influences the MRR. The carbon decomposed from the mineral oil contaminates the dielectric, resulting in an increase of abnormal discharge occurrence. The deionized water does not present such a problem. Therefore, the machining time of deep hole drilling using deionized water is much shorter that with mineral oil (Figure 6).

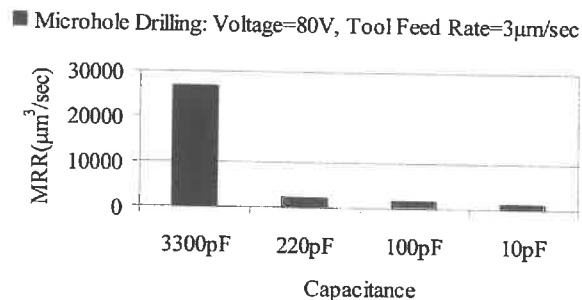


Figure 1 MRR vs. Capacitance.

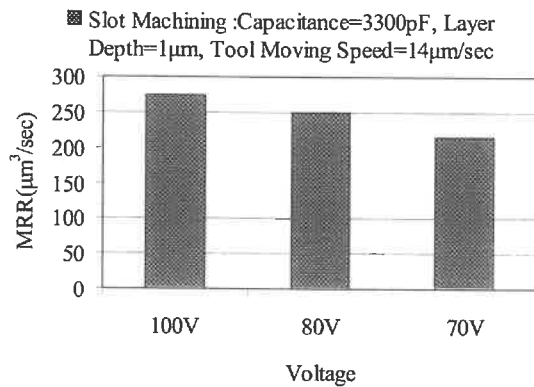


Figure 2 MRR vs. Voltage.

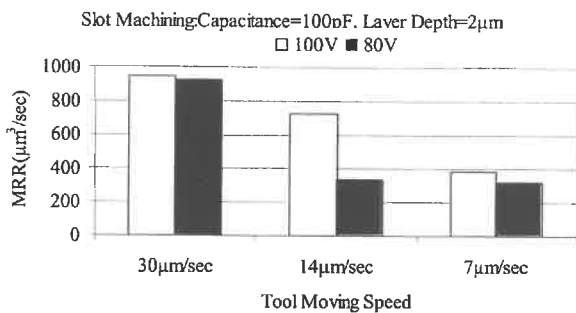


Figure 3 MRR vs. Tool moving speed.

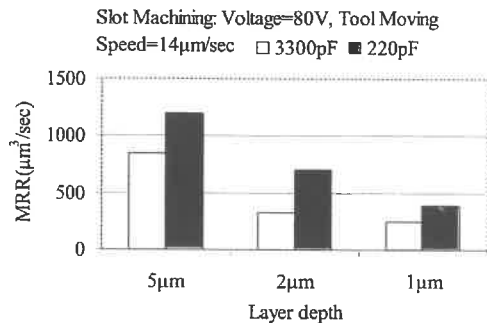


Figure 4 MRR vs. Layer depth.

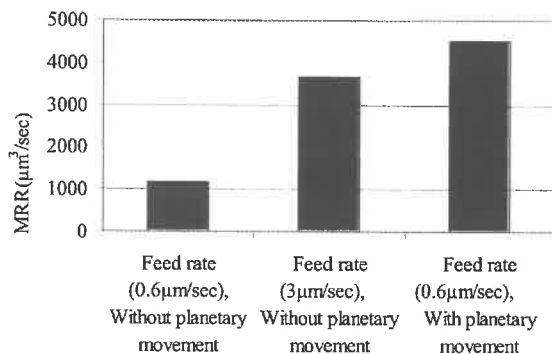


Figure 5 Effect of planetary movement on MRR.

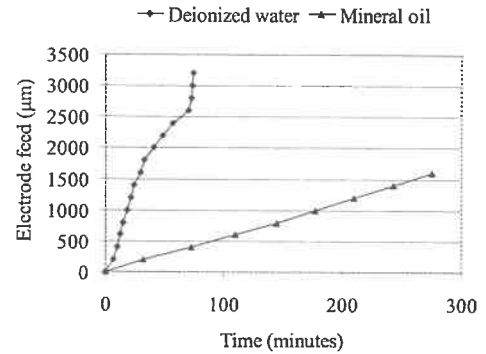


Figure 6 Effect of dielectric on Machining time.

Effect on EWR

The electrode tool wear ratio (EWR) is defined as the ratio between the electrode wear volume and the removed workpiece volume. Figures 7 and 8 show EWR increases with an increase of pulse energy (voltage and capacitance). In contouring Micro EDM, an increase of tool moving speed or layer depth causes the occurrence of abnormal discharge easily because the MRR is less than the tool moving speed. Figures 9 and 10 reflect this phenomenon under different conditions. In drilling of non-circular blind micro holes or deep hole by Micro EDM, the debris concentration occurs frequently, resulting in the abnormal discharges, followed by an increase of EWR. The planetary movement of electrode provides an extra space for an effective removal of debris and gaseous bubbles. Figure 11 shows the effect of planetary movement on EWR with an electrode nominal size was 50μm. It was found that the electrode area affects the electrode wear ratio in Micro EDM. The EWR reduces with the increase of electrode working area.

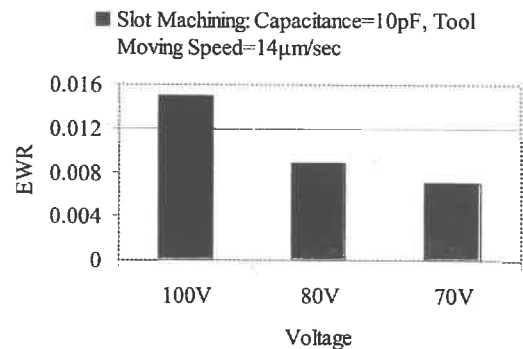


Figure 7 EWR vs. Voltage.

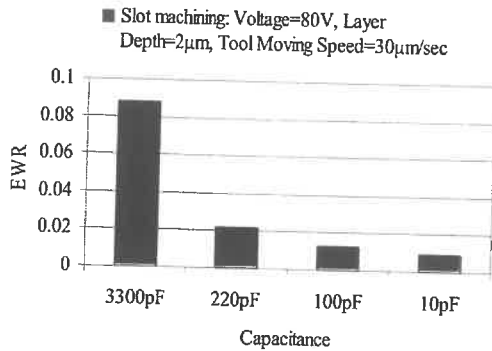


Figure 8 EWR vs. Capacitance.

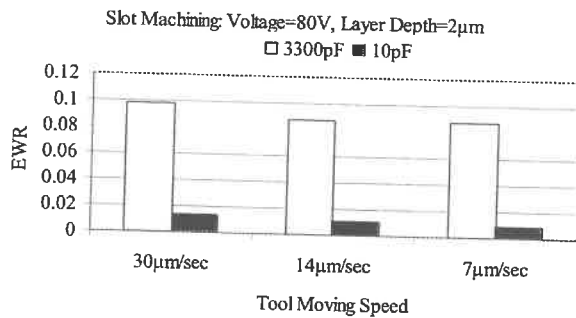


Figure 9 EWR vs. Tool Moving Speed.

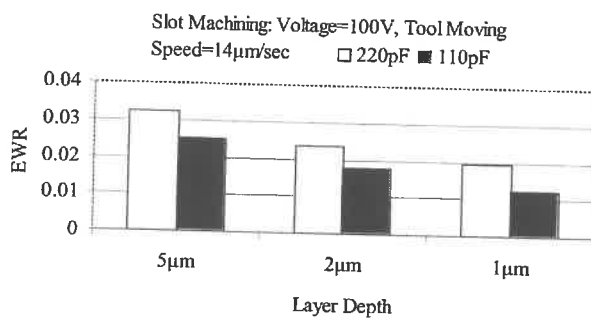


Figure 10 EWR vs. Layer Depth.

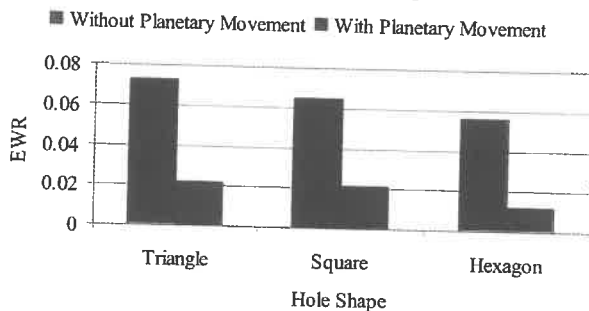


Figure 11 Effect of planetary movement on EWR.

Effect on the Discharge Gap

The discharge gap is calculated by dividing the difference between the width of slot and the diameter of the electrode by two. Figure 12 shows that the

discharge gap increases with an increase of pulse energy, i.e. voltage and capacitance.

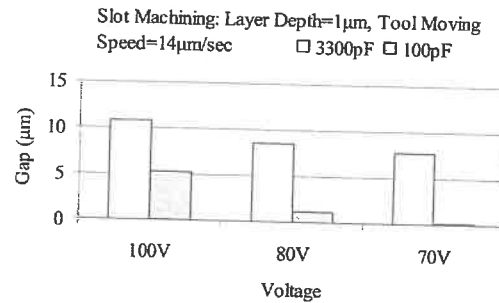


Figure 12 Gap vs. Voltage and Capacitance.

Effect on the Surface Roughness

The surface roughness increases with an increase of voltage and capacitance as shown in Figures 13 and 14. The craters on AISI 304 stainless steel appear circular and the size varies with the pulse energy as shown in Figure 15 (a) and (b). However, the craters on silicon are different from those on AISI 304 as shown Figure 15 (c) and (d) due to the different thermal behavior of two materials.

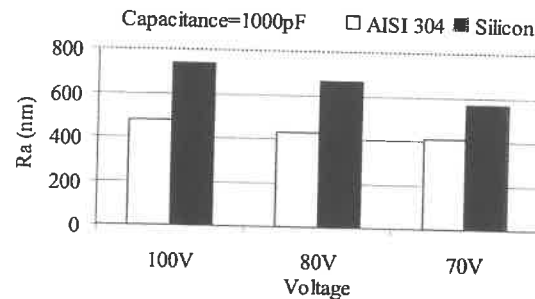


Figure 13 Ra vs. Voltage.

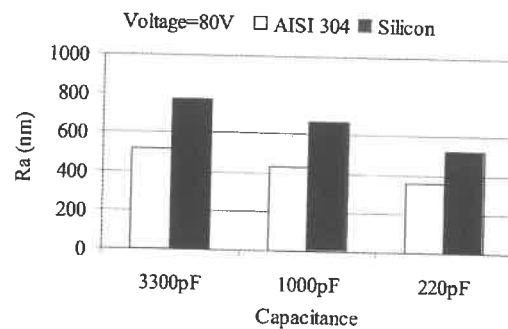
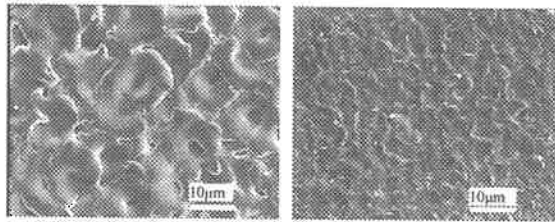


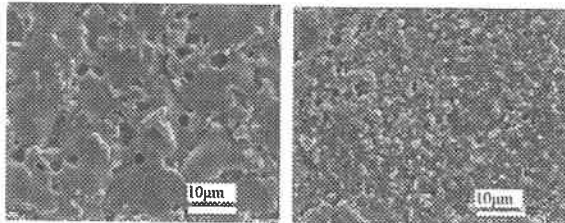
Figure 14 Ra. Vs. Capacitance.

Re-cast layer in Micro EDM

As Micro EDM is a thermal process, it generates a heat affected zone consisting of a re-cast layer and micro cracks. The thickness of re-cast layer in Figure 16 is around 10 μ m. The re-cast layer can be reduced by changing the tool path or layer depth, or can be removed using ECM or laser processes.



(a) AISI304, 80V, 3300pF. (b) AISI304, 80V, 220pF



(c) Si, 80V, 3300pF. (d) Si, 80V, 220pF.

Figure 15 Craters on AISI 304 and Si.

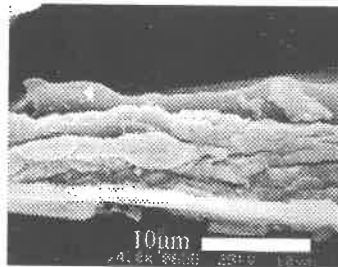


Figure 16 Re-cast layer in Micro EDM.

SOME EXAMPLES OF MICRO EDM

Figure 17 shows a cavity machined using a cylindrical electrode. The tool paths were generated using a CAD/CAM software. To maintain the tool tip shape unchanged and compensate the tool wear in length, the newly developed Uniform Wear Method was integrated with the tool paths generation [4].

A square cavity shown in Figure 18 was machined by applying the planetary movement to the electrode. Since the planetary movement of electrode avoids the debris concentration and reduces tool wear. Therefore, a micro cavity can be machined with sharp edges and corners. Based on the same working principle, a micro hole with a high aspect ratio can be drilled. Figure 19 shows a micro hole of 130µm in diameter through 1.8mm AISI 304L plate.

SUMMARY

This paper presents the effect of machining parameters on the Micro EDM process performance. It is clear that the machining performance has a close relationship with the pulse energy (voltage and capacitance). The material removal rate, wear ratio and surface roughness increase with the increase pulse energy. This paper illustrates that the

machining performance can also be affected by other factors such as aspect ratio, debris distribution, dielectric and tool path. When Micro EDM is employed to generate micro feature, part or mold, it is necessary to consider these factors for optimal process performance.

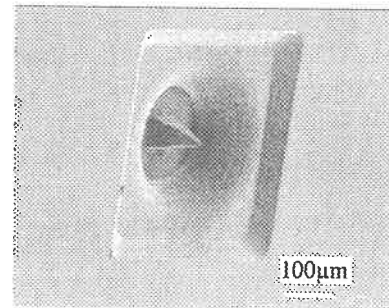


Figure 17 Micro cavity.

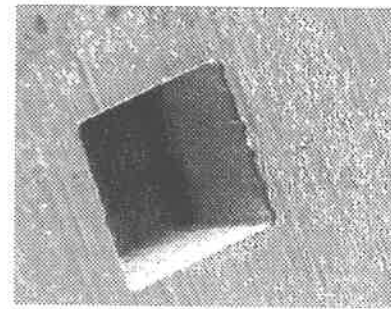


Figure 18 Square cavity.

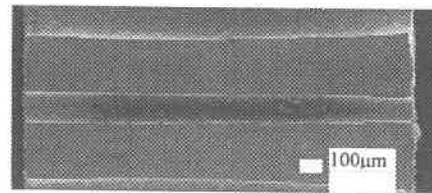


Figure 19 Micro deep hole.

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An Ultra-high Speed Spindle for Micro-milling

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1.0 Introduction

Milling is the fabrication technology of choice for a wide range of complex three-dimensional components, in a wide variety of materials. It is most commonly used to produce features ranging in size from a few millimeters up to thousands of millimeters. Small diameter milling cutters (~0.10 mm dia.) are widely and inexpensively available, and are mostly used to create small features in graphite EDM electrodes used to manufacture injection molding dies. There exists a wide variety of important applications for micro- and meso-scale mechanical systems which require high-strength materials and complex geometries that cannot be produced using current MEMS fabrication technologies. Micro-milling [9] is one fabrication technology with potential to fill this void by adding the capability of free form machining of complex 3D shapes from a wide variety and combination of engineering alloys, and other materials.

2.0 Difficulties and Challenges of Micro-Milling Technology

Micro-milling technology has many challenges associated with the scaling of the milling process, resulting in inefficiencies in the process. These include:

1. *Tool failure.* Micro milling tools often fail by fracture at the tool root, as opposed to edge wear. Because of the small diameter of the tools, the cutting forces must be kept very small so as not to exceed the bending stress fatigue limit of the tool.
2. *Chip Load:* In order to keep the cutting forces sufficiently small, the chip thickness must be very small, often less than 1 micrometer.
3. *Cutting Edge Radius:* Commercially available micro-end mills typically have cutting edge radii on the order of 2 to 3 micrometers, substantially higher than the desired chip thickness. This means that the effective rake angle of the cutting edge is highly negative, leading to cutting force coefficients much higher than in conventional milling..
4. *Spindle Runout:* Commercially available micro-milling spindles typically have radial runout at the tool tip on the order of 1 to 2 micrometers or more, larger than the desired chip thickness; leading to overloading of the tool and premature failure.
5. *Cutting Speed:* As the tool diameter decreases, the spindle speed must increase to achieve the same (optimum) peripheral cutting speed. For very small tools, spindle speeds over 500,000 rpm are required.

Inefficiencies in milling small features result directly from the relationships between a cutting tool's breaking strength, the cutting force, and the material removal rate (MRR). End mills act much like long, thin, end-loaded cylindrical cantilever beams, where the maximum bending stress (σ_x) is:

$$\sigma_x = \frac{32 F_c L}{\pi D^3} \quad (1)$$

where L is the tool length, D is the tool diameter, and F_c is the transverse cutting force. For a given L/D aspect ratio of the tool, the bending stress increases with the square of reductions in tool diameter. Therefore, in order to prevent tool breakage, the cutting force must be reduced. The maximum tangential cutting force is approximately:

$$F_c (N) = K_s (N/mm^2) \left[b (mm) \left(\underbrace{\frac{f_r (mm/min)}{n (rev/min) m (teeth/rev)}}_{ChipLoad} \right) \right] \quad (2)$$

where K_s is the cutting force coefficient of the workpiece material, b is the axial depth of cut, and f_r is the feedrate, n is the spindle speed, and m is the number of teeth on the cutter. To reduce the cutting force, it is necessary to reduce

the chip load, or maximum chip thickness. This can be accomplished either by decreasing the feed rate, or increasing the spindle speed. For a cut with radial depth, a , the MRR is:

$$MRR \left(\frac{\text{mm}^3}{\text{min}} \right) = f_r \left(\frac{\text{mm}}{\text{min}} \right) b(\text{mm}) a(\text{mm}) \quad (3)$$

From Equations 1-3, we can see that as we reduce component size by a scaling factor, $k < 1$, the MRR decreases at the rate of k^3 . If spindle speed is held constant, the cutting force must reduce by a factor of k^2 , necessitating a reduction in the feed rate by a factor k . This relationship does not take into account the large tool edge radius and spindle/tool runout relative to the chip thickness in micro-milling, which causes higher cutting force coefficients and tends to increase cutting forces for a given chip area, necessitating even further reductions in feed rate to prevent tool breakage. One solution to this problem is to increase spindle speed to maintain the desired peripheral cutting speed, while also minimizing runout. However, for very small diameter end mills, the required spindle speeds can be 500,000 rpm or higher.

3.0 Ultra-high Speed Micro-milling Spindle Design Concept

The previous analysis of micro milling demonstrates the need to develop a spindle capable of extremely high rotational speeds, while maintaining sub-micron radial error motions. Commercially available micro-milling cutters typically have a 0.125-inch diameter shank, approximately 1.25 inches long, with the actual cutting flutes ground on the end of the shank. The proposed spindle system supports the tool shank in a porous carbon air bearing, custom designed for this application. The tool shank itself is driven by a friction wheel which is driven by a commercial high speed spindle. The drive ratio of the friction drive system is approximately 8:1, making it possible to achieve tool speeds over 500 krpm using commercially available high speed motorized spindles. The friction wheel is coated with a compliant polymer coating which reduces the contact stiffness, and helps to insure that error motions of the drive spindle will not be transmitted to the tool shank. The drive spindle is mounted on the base frame with a set of spherical washers and 4 adjusting screws. This arrangement allows the motor axis to be tilted in two perpendicular directions to control the normal force between the friction wheel and tool shank. The drive spindle axis is slightly tilted relative to the tool shank axis, so that the friction force has a small vertical component, forcing the tool shank against the thrust air-bearing stop. This also allows easy tool change by stopping the motor and rotating the tool shank by hand in the opposite direction to force the tool out of the air bearing. The entire air bearing assembly is supported on a 3-axis force sensor to allow for continuous measurement of cutting forces. The conceptual design of the ultra-high speed micro milling spindle is shown in Figure 1.

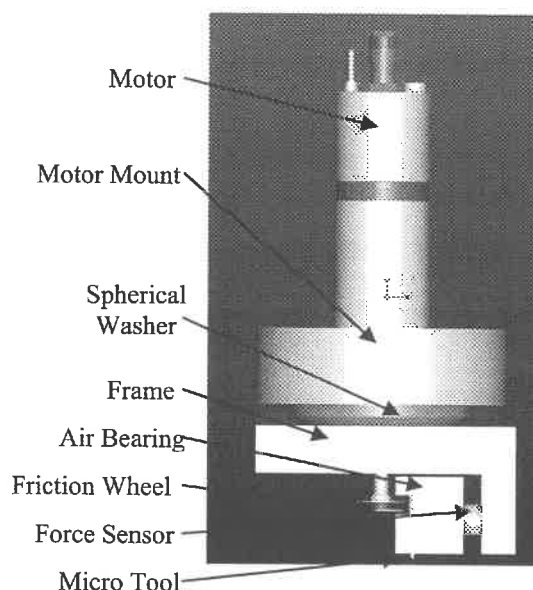


Figure 1. Spindle conceptual design

3.1 Model Development & Analysis

The drive spindle is selected based on torque and power calculations using Tansel's [1] cutting force model for micro-milling, and the projected tool and material combinations. A finite element analysis of the selected motor and friction wheel assembly predicted a lowest critical speed of 85k rpm with aluminum as the friction wheel material.

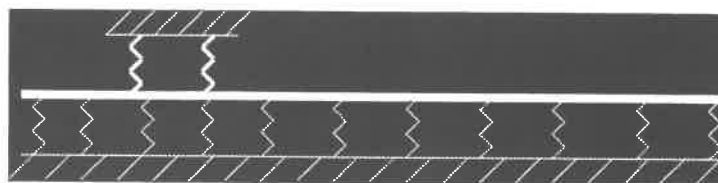


Figure 2: FE Model of the tool air-bearing and friction wheel

A finite element analysis of the micro-tool/air-bearing/friction drive assembly was also performed. Figure 2 shows the FE model. The tool inside the air-bearing is modeled as a continuous beam supported on springs with composite stiffness equivalent to the air bearing stiffness. The stiffness of the custom air-bearing was predicted to be approximately 4.3×10^4 N/m by extrapolating from published stiffness values of other air bearings produced by the

vendor. The beam is also contacted by another spring with stiffness equivalent to the Hertzian contact stiffness between the tool and the friction wheel, which was estimated to be 1.68×10^6 N/m. The analysis predicts the lowest natural frequency of the system to be 4291 Hz. This corresponds to a rotational speed of approximately 257,000 rpm, and points to the need to increase the air bearing stiffness in future versions.

4.0 Experimental Set-up & Testing

The prototype micro-spindle was assembled and tested at the University of Florida Machine Tool Research Center. The nominal speed ratio between the drive spindle and the tool is 7.9:1 if no slip occurs. The micro-spindle system was interfaced with a National Instruments data acquisition card using LabView to allow measurement and control of the drive spindle speed, as well as 3-axis cutting force measurements, and run-out measurements. The air bearing is supplied with very clean air at 100 psi. For initial cutting tests, the micro-spindle was mounted on a large high speed milling machine which has hydrostatic guide-ways to insure low vibration levels in the machine. Figure 3 shows a photograph of the set-up.

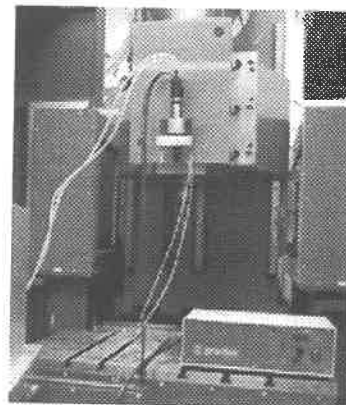


Figure 3: Experimental Set-up

4.1 Run-Out Measurements

Radial run-out measurements were carried out using Lion Precision capacitance probes. Initially, the radial runout of a point on the drive spindle arbor was measured at various spindle speeds both with and without the friction wheel mounted. The results are shown in Figure 4. There is a clear parabolic increase in radial runout when the friction wheel is mounted, which is attributed to centrifugal forces arising from wheel unbalance.

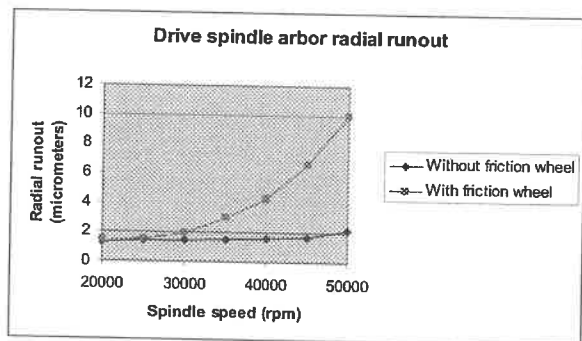


Figure 4. Run-out of drive spindle arbor

surface. These results show that the friction wheel surface is far too irregular for this application, and the imperfections in its surface are transmitted into the tool shank and amplified at the tool tip. Further analysis of the air bearing and friction wheel, and improved manufacturing methods for the friction wheel are needed to rectify this problem.

4.2 Cutting Force Measurements

Although the tool tip run-out was exceedingly high, preliminary cutting tests were performed to evaluate the cutting performance of the micro spindle. The initial cutting tests were performed at a drive spindle speed of 10,000 rpm which yielded a nominal tool speed of approximately 79,000 rpm. Slotting cuts were performed using tool diameters ranging from 0.03(762 μ m) inches to 0.005(127 μ m) inches. Slot depths of one-sixth of the tool diameter, and one third of the tool diameter were used. Figure 6 shows the feed direction force record for 127 μ m cutter with a 42.33 μ m slot depth and the largest achievable feed rate without tool breakage of 0.045 mm/min (0.376 μ m/tooth). The

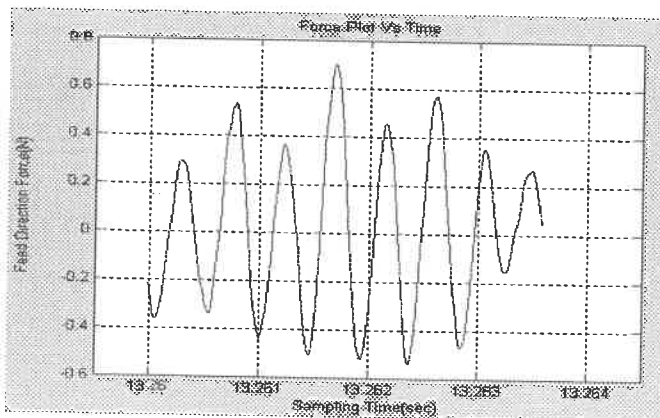


Figure 6: Feed direction force plot

passage of individual teeth is clearly evident in the force record. Significant variation in the peak cutting force is evident, and may be due to asynchronous error motions of the tool. Typical cutting test results are shown in Table 1. Both Thusty's [2] and Tansel's [1] cutting force models were used to calculate the cutting force coefficient, K_s , using the average maximum feed direction force. For

Tool Diam. (μm)	Width of cut (μm)	Depth of cut (μm)	Feed /tooth (μm)	Avg Max Force (N)	K_s (Thusty) (N/mm ²)	K_s (Tansel) (N/mm ²)
762	762	127	1.18	0.5072	5780	3340
762	762	127	1.66	1.6709	13500	7810
508	508	169.33	2.24	0.8575	3870	4470
508	508	169.33	2.63	1.195	4590	5300
254	254	84.66	0.479	0.2106	9690	11200
254	254	84.66	0.479	0.2101	7320	8470
127	127	42.33	0.187	0.1919	41400	47800
127	127	42.33	0.376	0.2421	26000	30100

Table 1: Cutting test results – Aluminum 6061

milling with macro tools, K_s for this material will be on the order of 800 N/mm². These results illustrate the increased specific cutting force coefficient as the chip thickness gets smaller than the edge radius of the tool. However, the large tool point runout in the current spindle makes the chip thickness vary for each tooth, and creates a large uncertainty in the computed cutting force coefficient.

5. Discussion and Conclusions

The design, assembly and initial testing of a prototype ultra-high-speed micro-milling spindle have been completed. The results are encouraging but significant challenges remain. The friction wheel shows excessive runout, which will necessitate proper balancing and improved methods for applying the friction coating. At drive spindle speeds greater than 20k rpm it appears that the tool shank begins to contact the inner surface of the air bearing. This is believed to be related to the problems with friction wheel runout and unbalance. Solution to this problem will require a more compliant coating on the friction wheel and improvements in the air bearing to increase its stiffness. Nonetheless, the spindle is capable of operating at speeds in excess of 500,000 rpm for short periods. Recorded cutting forces are comparable to those reported by other researchers.

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Fundamental Analysis on the Novel 3-D Probing Technique for Microparts Using the Optical Fiber Trapping

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A novel three-dimensional probing technique for microparts with high aspect ratios is proposed. In this technique, a particle trapped optically at the optical fiber tip is used as a probe for position detection. In order to confirm the feasibility of this method, this paper demonstrates numerical computations about the optical force exerted on the probe sphere and the behavior of position detection signal using finite-difference time-domain method and Maxwell stress tensor. It is found that the optical force induced at the tapered fiber tip can trap the probe sphere three-dimensionally, and it has the performance as the position detection probe.

Key words: CMM, high aspect ratio, fiber trapping, finite-difference time-domain method

1. Introduction

With the recent development of micro electro mechanical systems (MEMS), mechanical parts have been smaller in the size of micrometer order. And using a variety of micro fabrication methods such as the photolithography, the LIGA process, etc., we can fabricate micron-sized three-dimensional (3-D) shapes. To guarantee the accuracy of microparts, it is indispensable to measure 3-D profiles like sizes and positions. At present, however, it is impossible to evaluate the geometrical quantities three-dimensionally. The 3-D coordinate measuring machine (CMM) for microparts with the measuring range of micrometers and the accuracy of nanometers is required. We reported the possibility of the probing technique using the laser trapping technique [1-3]. However, this technique can't be applied to micro objects with high aspect ratios such as micro deep holes and grooves because of the shadow effect. In truth, the measuring system that enables to measure the micro shapes with high aspect ratios has not been realized yet.

This paper proposes a novel 3-D probing technique for measuring microparts with high aspect ratio. This technique uses the optical force exerted on a dielectric particle in the intensity gradient field of light [2]. A particle with a diameter of 5 ~ 10 μm trapped optically at the tip of an optical fiber plays the role of a probe for position detection, and the fiber with a diameter of smaller than that of the particle works as a stylus shaft.

The proposed probing scheme requires to trap a particle with 5 ~ 10 μm diameter against the gravity in the air using the light emitted from an optical fiber tip. There are many studies on the trapping technique using optical fibers and a nanometric aperture. These optical trapping techniques were derived from the work of Ashkin and co-workers on the optical trapping technique [2,3]. There have been various interesting studies on the fiber trapping technique, however, these used "two or more optical fibers in water" [4-6]. Okamoto and Kawata theoretically analyzed that an evanescent field near a nanometric aperture of metallic layer could trap subwavelength particles "in water" [7]. Choi et al. demonstrated similar calculations for nanometric particles [8]. Novotny et al. and Chaumet et al. showed theoretically the trapping potential for nanoparticles in the field scattered by "a metal tip" [9,10]. The configurations of these studies are unavailable for our probing technique. Therefore, we have to confirm the possibility of the fiber trapping technique as used in our proposed method.

In this report, to validate our proposed method, we analyze numerically the optical force exerted on the probe sphere illuminated by the beam emitted from the fiber tip using finite-difference time-domain (FDTD) method and Maxwell stress tensor.

2. Principle of position detection

Fig. 1 shows the principle of the fiber trapping probing technique. A dielectric particle with a diameter of 5 ~ 10 μm is trapped by the optical force induced by the illumination light from the fiber tip. The particle serves as a positional detection probe and the optical fiber works as a stylus shaft. The optical fiber consists of the main body part, which is a single mode optical fiber, and the micro optical fiber part, whose core is exposed to the air. At the micro optical fiber part, the air serves as its cladding. The micro optical fiber part is thinner than the diameter of the probe sphere and longer than the depth of shapes like deep holes and grooves. This enables the probe to make an approach to the steep angle surface of the workpiece with high aspect ratios. The incident beam propagates through the fiber and is emitted from the fiber tip. The light traps the probe sphere optically, and is scattered by the spherical probe surface. Then, the scattering light returns into the fiber tip again and guided to the other fiber end, and its intensity is

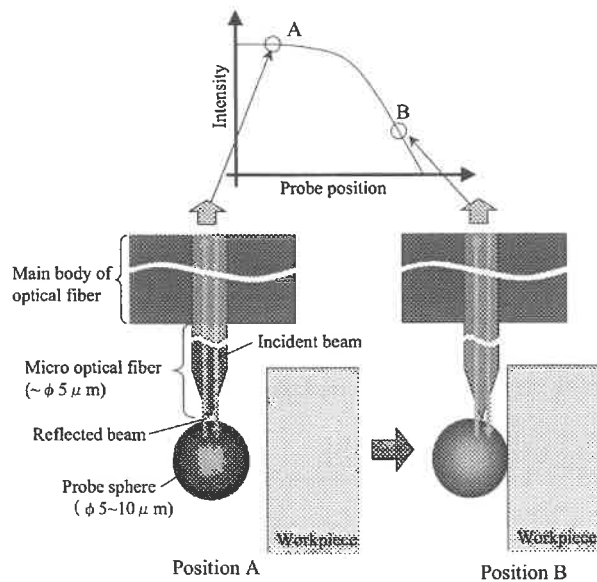


Fig. 1 Principle of position detection

lyzed region is divided into discrete grids of points, and the finite-difference representation of Maxwell's curl equations using a central-difference formulation and the Yee-cell notation is applied to each grid. This computational method is widely and effectively used to solve electromagnetic problems with complicated configurations.

Once the scattering electromagnetic field is determined, we can obtain the optical force from the electromagnetic field by the Maxwell stress tensor [8-11]. Optical force is a kind of electromagnetic force. According to the conservation of momentum, providing momentum to an object generates the force acting on the object. In the case that the object receives the electromagnetic momentum from the light, optical force is exerted on the object. The electromagnetic momentum is expressed as Maxwell stress tensor given by

$$\mathbf{T} = \epsilon_0 \epsilon \mathbf{E} \mathbf{E} + \mu_0 \mu \mathbf{H} \mathbf{H} - \frac{1}{2} \mathbf{I} (\epsilon_0 \epsilon \mathbf{E} \cdot \mathbf{E} + \mu_0 \mu \mathbf{H} \cdot \mathbf{H}) \quad (1)$$

where ϵ_0 and μ_0 are the electric permittivity and the magnetic permeability of vacuum, and ϵ and μ stand for the dielectric constant and magnetic permeability of surrounding medium, respectively. By integrating this Maxwell stress tensor on the surface surrounding the object, the change of the momentum inside of the surface can be found. Since the momentum changes on the boundary of the object, the extinction amount of the momentum becomes the force acting on the object. The electromagnetic force $\mathbf{F}$ in a steady state is given by,

$$\mathbf{F} = \iint_S \langle \mathbf{T} \cdot \mathbf{n} \rangle dS \quad (2)$$

where $\langle \dots \rangle$ represents the time average, and $\mathbf{n}$ is the outwardly directed normal unit vector.

Fig. 2 shows the geometry of the model for the numerical analysis. The analysis region of $12 \mu\text{m} \times 7.4 \mu\text{m} \times 14 \mu\text{m}$, which is the foremost part including the fiber tip and the spherical probe in the air, is considered. The origin of the coordinate system (x, y, z) is located at the fiber tip, and the top of the probe sphere P represents its position. The probe is a silica sphere ($n=1.44$) with diameter $5 \mu\text{m}$. The micro optical fiber surrounded by the air cladding ($n = 1.0$) has $4 \mu\text{m}$ core ($n = 1.47$) diameter. Nd:YAG laser ($\lambda = 1064 \text{ nm}$) is guided by the fiber, and emitted from its tip which is tapered with tip radius $r = 500 \text{ nm}$ and tip angle $\alpha = 45^\circ$. In practice, this micro fiber with air cladding is a multimode optical fiber at $\lambda = 1064 \text{ nm}$, and many modes propagate in it. However, the HE_{11} mode in the main body part couples the HE_{11} mode in the micro fiber part with 69% coupling efficiency at the boundary between the main body part and the micro fiber part, and the other light

monitored at the fiber extremity. In the measurement, if the probe sphere moves by touching the measured surface, the scattered light changes with the displacement of the probe sphere. In this way, the shift of the particle can be detected by monitoring its intensity at the other fiber extremity.

3. Theory and numerical modeling

In this report, we analyze numerically the optical force exerted on the particle illuminated by the laser light from the fiber. Optical forces are the result of the transfer of momentum from the illumination light to the particle in the process of light scattering. This transfer of momentum can be obtained by solving the electromagnetic scattering problem. Thus, it is essential to calculate the electromagnetic field that is scattered by the optical fiber tip and the particle. Solving an electromagnetic scattering problem requires the solution of the Maxwell equations. We employ the 3-D FDTD method to obtain the solution of the Maxwell equations [12]. In the FDTD method, the analyzed region is divided into discrete grids of points, and the finite-difference representation of Maxwell's curl equations using a central-difference formulation and the Yee-cell notation is applied to each grid. This computational method is widely and effectively used to solve electromagnetic problems with complicated configurations.

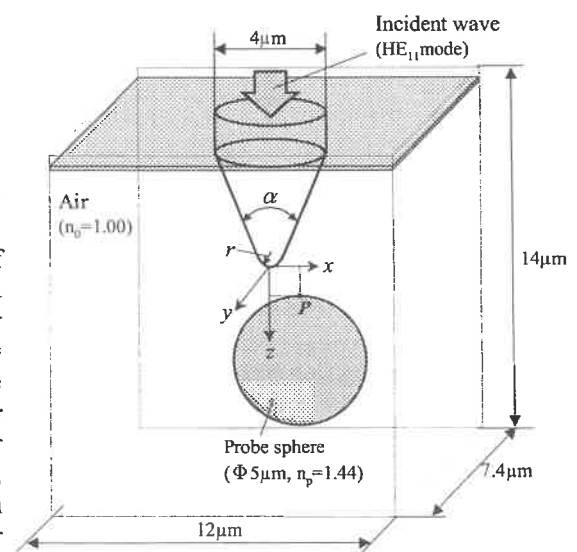


Fig.2 The geometry of the model. The fiber has a tip radius $r = 500 \text{ nm}$ and a tip angle $\alpha = 45^\circ$.

leaks at the boundary or couples other modes. Since most light power is provided to the HE_{11} mode in the micro fiber, we assume that only HE_{11} mode is guided to the fiber tip and employ the HE_{11} mode as the incident wave.

We compute the electromagnetic field of this model using FDTD method, and calculate the trapping force exerted on the spherical probe by applying Eq. (1) and (2) to the cubic surface surrounding the probe. We take the time average of the force over the period of the electromagnetic field and continue these successive calculations until the force values no longer change significantly. In order to find the relation of the position of the probe and the force, the probe position was changed near the tip and the successive calculations are performed for each position.

4. Optical force exerted on probe sphere

Fig. 3 shows the electromagnetic field distribution near the fiber tip in the x - z plane. The spherical probe is absent in Fig. 3 (a), and is placed under the fiber tip in Fig. 3 (b). As shown in Fig. 3(a), the tapered structure of the fiber focuses the light to the tip and generates an evanescent field in the vicinity of the tip as the illumination mode fiber of NSOM. In Fig. 3(b), it is shown that the probe scatters the evanescent wave. This change of the electromagnetic field on the probe surface gives rise to the optical force acting on the probe sphere.

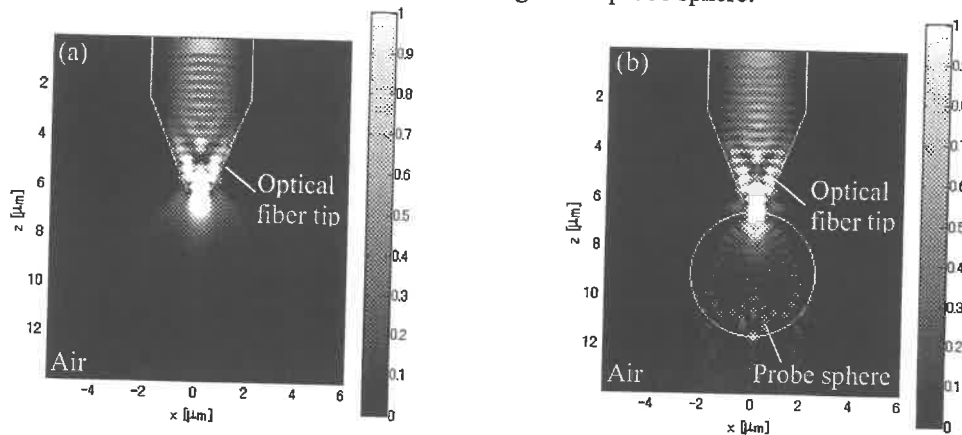


Fig.3 The distribution of the electromagnetic field in the vicinity of optical fiber tip (a) without the probe sphere, (b) with the probe sphere

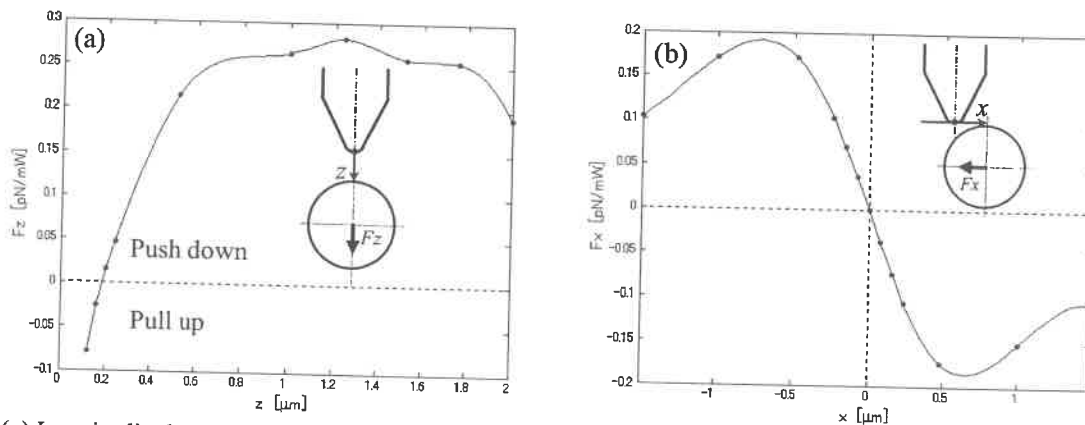


Fig.4 (a) Longitudinal optical force and (b) transverse optical force, in pN per milliwatt of total power transmitted in the optical fiber.

The calculated optical force for each probe position is shown in Fig. 4. Fig. 4(a) shows the longitudinal optical force acting on the probe sphere located along the fiber axis. A positive force acts to push the probe down, and a negative force acts to pull it up. We note that the pushing force acts on the probe in the region of $z > 0.2 \mu\text{m}$, and pulling force acts on it in the region of $z < 0.2 \mu\text{m}$. Fig. 4(b) shows the evolution of the transverse force acting on the probe as the probe moves laterally. The result indicates that when the probe shifts its position, the optical force acts to attract the probe to the fiber axis. In both the direction along the z and x axes, the probe is attracted toward the strong intensity field. This phenomenon is similar to the conventional optical gradient trap in which a particle is trapped near a spot focused by an objective lens. The weight of the silica particle as probe (density 2.0 g/cm^3) is 0.13

pg. Since, in the vicinity of the tip, the pulling up force of 0.078 pN/mW exerts on the probe sphere, we can trap the probe sphere at the tip the fiber tip against the gravitational force with 16.4 mW transmission light power. In the region of $-0.5 < x < 0.5$, there is a linear relation between the displacement and the transverse force. Therefore, the spring constant of this probe system can be found. In the case of 16.4 mW power, we obtain the lateral axial spring constant $k = 5.8 \times 10^{-6}$ N/m, which is much lower than the spring constant of the AFM ($k = 0.1$). Consequently, this probe can have quite a low measuring force and avoid the damage to delicate samples.

5. Position detection signal

In order to characterize the behavior of position detection signal, we analyzed the reflection light signal returning into the fiber while the probe is approaching the measured surface. Fig. 5 shows the simulation result. The horizontal axis indicates the spatial relation x of the measured surface and the approaching point O on the probe at the equilibrium position. The positive x means the distance between the approaching point O and the measured surface, and the negative x means the displacement of the probe (see Fig. 5). The vertical axis indicates the intensity of the detection signal that is calculated by integrating the electromagnetic field intensity of the reflected wave inside of the fiber. We obtained the electromagnetic field of the reflected wave by subtracting that of the incident wave from the total electromagnetic field inside of the fiber. We note that when x is positive, there is no change of the detection signal. When the probe contacts the measured surface and the probe shift becomes more than 200 nm, the signal intensity increase with the shift amount. Then if the shift amount exceeds 300nm, the signal intensity decays substantially. We can detect the position of the measured surface by sensing this substantial change of the signal. However, it is a problem that the probe needs such a large shift of 200 ~ 300 nm until the position is detected. In the actual measurement, we have to calibrate this bias value. It is indispensable to improve the system for more sensitive probe.

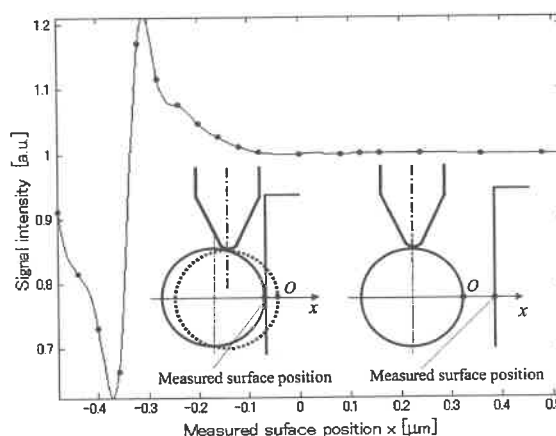


Fig.5 The intensity of position detection signal and displacement of the measured surface position x .

6. Conclusion

The fundamental analysis on the fiber trapping probe technique is demonstrated using FDTD method and Maxwell stress tensor. We confirmed that the probe sphere could be trapped three-dimensionally by optical force from a tapered fiber tip with a tip radius 500nm and a tip angle 45° . The lateral spring constant of the probe is 5.8×10^{-6} N/m, which is much lower than the spring constant of the AFM. Therefore, this probe can have quite a low measuring force and avoid the damage to delicate samples. In addition, we characterized the position detection signal. The probe shift beyond 200 ~ 300 nm gives rise to the substantial change of the signal intensity. It is possible to detect the position of the measured surface by sensing the signal change. However, it is necessary to improve the probing system for the more sensitive probe.

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Rotary Type Micro-Engineered Tool

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Abstract

The disk is with ultrasonic excitation in the radial direction. Laser-machined cutting edges are aligned to have a sinusoidal phase shift corresponding to successive grit point spacing. The sinusoidal phase shift is designed and calculated so that single cutting edge has the depth of cut below critical depth of cut and intermittent cutting is attained. As a result, single cutting realize shear-mode step-wise removal of brittle materials, tool depth of cut is several hundred times of critical depth of cut, that is proportional to the numbers of cutting edges. At the same time, intermittent cutting is expected to enable shorter edge contact period with materials and eventually minimize cutting edge wear.

1 Introduction

We developed and reported an micro-engineered diamond tool to perform high -quality shear-mode machining of brittle materials. It was found that the tool realize very large tool depth of cut compared conventional fine grit grinding tools and diamond cutting tools. However, the use of the tool was limited to the machining of flat surface and convex surface with no inflection points. Rotary type micro-engineered tool to be described here, can be applied to the machining of any type of surfaces such as aspherers.

2 An "Successive grit Point Spacing Model" and the principle of Rotary Type Micro-Engineered Tool

Ultrasonic cup-wheel grinding showed almost constant grinding force and smooth surface with roughness in nanometers. 1) The removal mechanism shown in Fig. 1 is supposed to be explained by the following sinusoidal motion of grits.

The successive grit point spacing (p) is give by $P(=w \cdot 2/b)$

Here, w is the average grit point spacing and b is the average width of the groove.

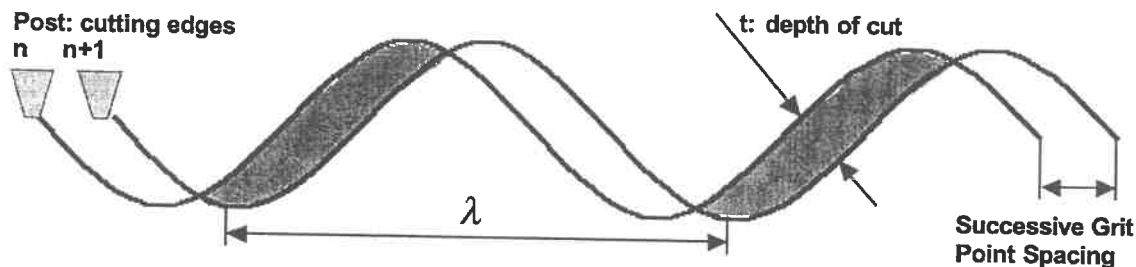


Fig.1 Successive grit motions with vibration-excitation

When the wheel is excited with ultrasonic vibration of frequency f and amplitude a , the motion of grits can be expressed by a sinusoidal function and each grit has a phase shift of (P)

$$f(x) = a/2 \cdot \sin[(2\pi/\lambda) \cdot (x - n \cdot p)]$$

Here, λ is the wave length.

Thus distributed grits are presumed to remove material at a depth lower than the critical depth of cut under certain machining conditions depending on feed velocity, ultrasonic frequency and amplitude.

Figure 2 shows the principle of Rotary type micro-engineered tool.

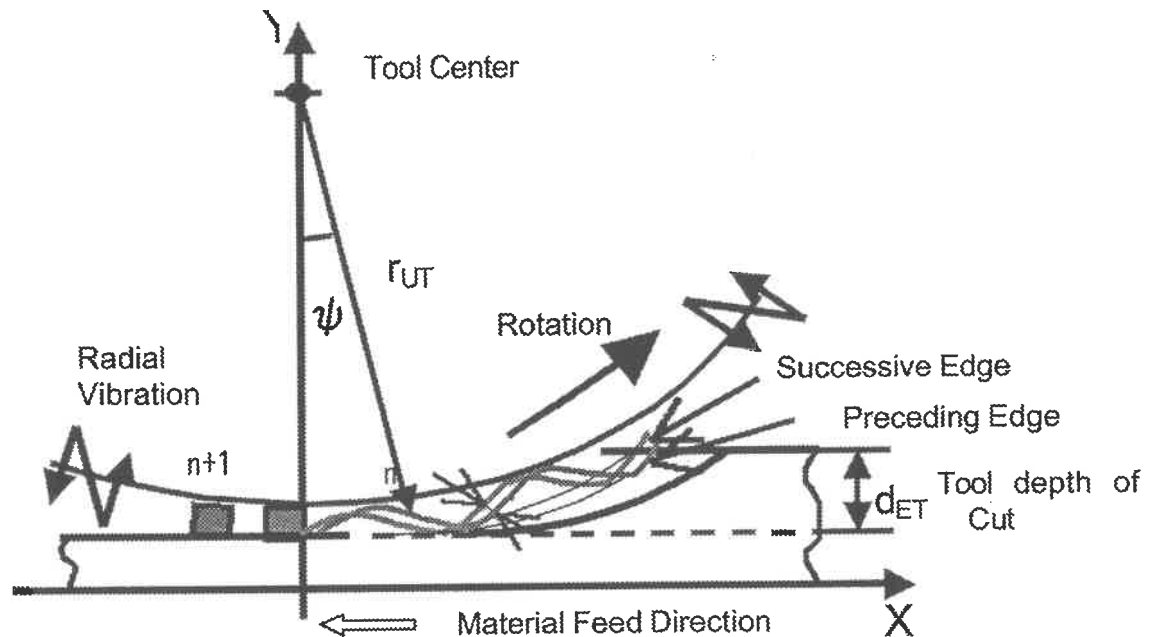


Fig. 2 Rotary Type Engineered Tool

Multi-cutting edges are arranged on the peripheral surface of a disk. The disk is rotated with ultrasonic excitation in the radial direction. When the disk is excited in the radial direction, the cutting edge radius r_{UT} of micro-engineered tool is expressed as,

$$r_{UT} = r_T + a \sin \phi$$

where r_T is the radius of the cutting edge, a is the amplitude and ϕ is the phase.

Suppose that the material feed rate, V_f , tool rotation, N , rotation angle, Ψ , and cutting edge motion are expressed as

$$X_{post} = \frac{V_f}{2\pi N} + r_{UT} \sin \Psi$$

$$Y_{post} = r_T - r_{UT} \cos \Psi$$

The sinusoidal phase shift can be calculated so that a single cutting edge has a depth of cut lower than the critical depth of cut and intermittent cutting is attained. As a result, a single cutting can realize shear-mode step-wise removal of brittle materials, tool depth of cut is several hundred times the critical depth of cut which is proportional to the number of laser

machined cutting edges. Moreover, intermittent cutting is expected to enable a shorter edge contact period with materials, minimizing cutting edge wear.

3 Expected micro-removal mechanisms

Cutting edges draw combined sinusoidal and trochoidal motions as shown in Fig.3.

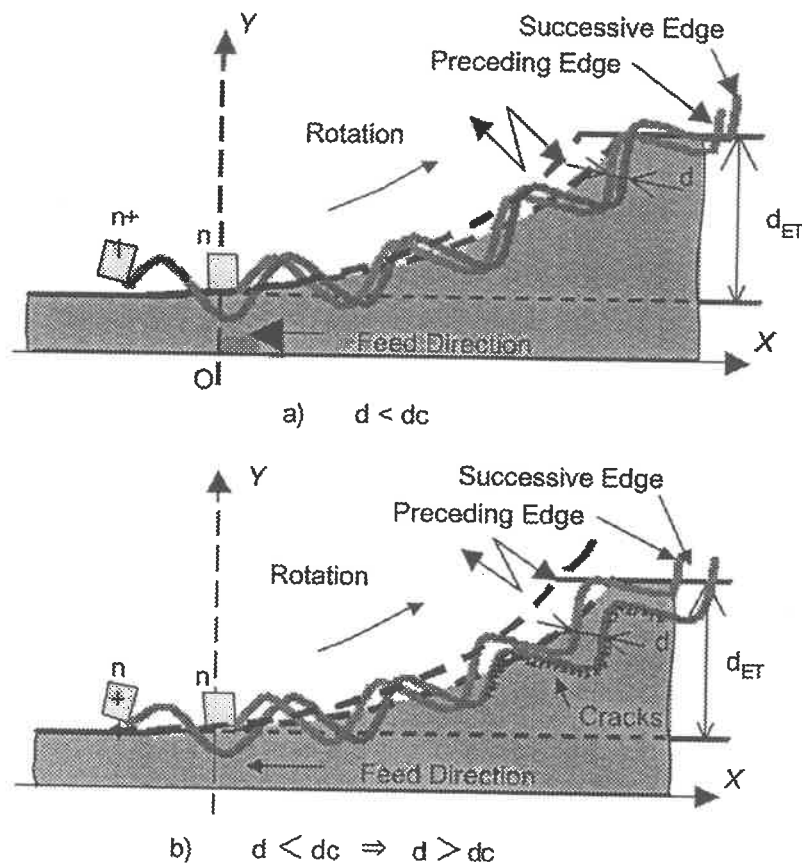
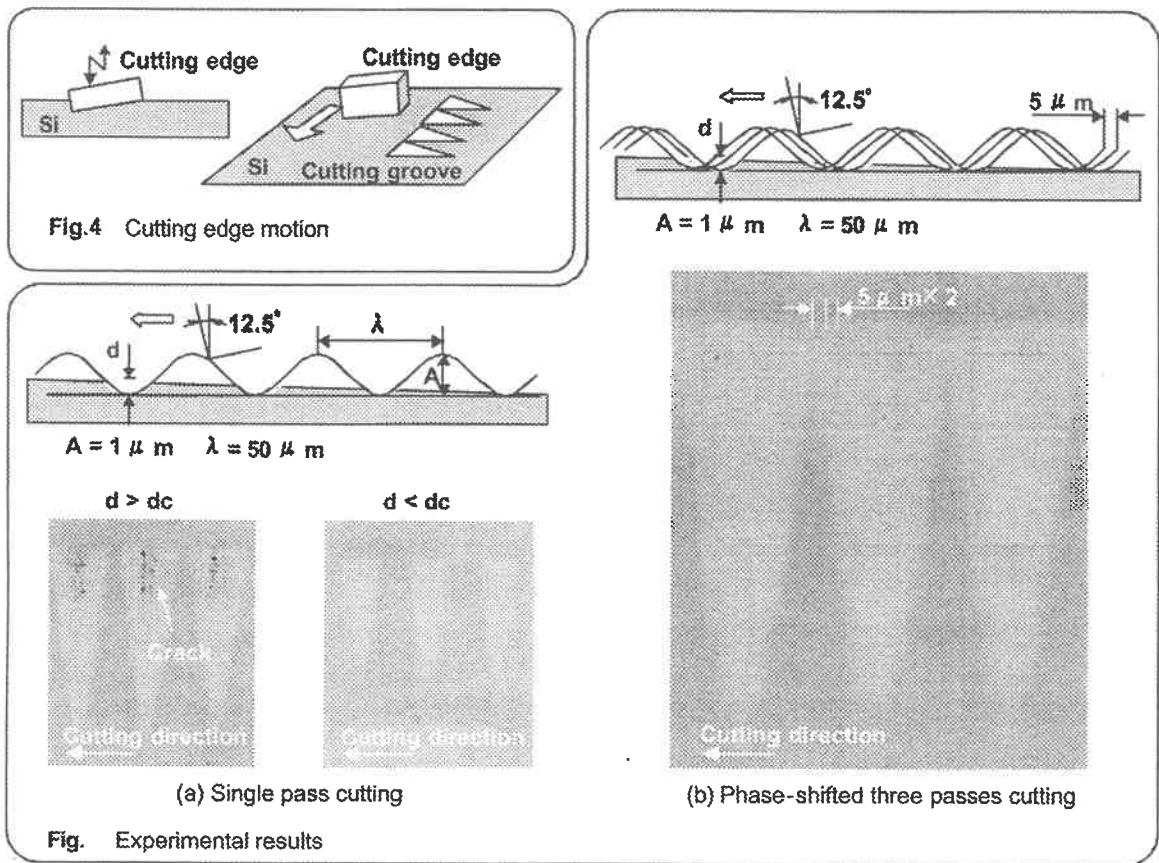


Fig.3 Micro removal mechanism

Conditions under $d < d_c$ in the range of smaller d_{ET} , machining is realized in all shear-mode as shown in Fig.3 a).

When the total depth of cut d_{ET} is increased, material is removed in brittle-mode on the upper surface of material and in shear-mode near the surface to be left as finished surface as shown in Fig.3 b). Then the shear-mode machining with a large d_{ET} is to be expected in a fairly low machining force.

4 Cutting experiments of Si on the nanometer-resolution machine tool, Robo-nano, produced by Fanuc



5 Conclusions

The principle of rotary type multi-edge diamond tools were investigated to attain the higher productivity in shear-mode machining of brittle materials. Further, some experiments were conducted.

The features of the micro-engineered tools are;

- 1) It was found that tremendous cutting edges with vibration excitation make it possible to give a large depth of cut at a time, while each cutting edge removes materials under critical depth of cut.
- 2) Designing cutting edges to have the phase shift one next to each other, Cutting edge removes materials intermittently. This is expected to enable a shorter edge contact period with materials, minimizing cutting edge wear.
- 3) Cutting experiments of Si showed material removal in shear-mode when the depth of cut is lower than the critical depth cut $0.2 \mu\text{m}$.
- 4) It was found that the phase shift of successive edge motion does not cause any defects such as brittle fractures.

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FABRICATION OF A PRECISION MANDREL FOR REPLICATING WÖLTER X-RAY OPTICS

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Lawrence Livermore National Laboratory

1. Introduction

With the constant push to miniaturize existing technologies, there is an ever-increasing need to characterize smaller and smaller objects. X-rays have proven their usefulness for characterizing the internal structure of objects. However, standard x-ray imaging (i.e. projection radiography) methods are not ideally suited for high-resolution imaging of small objects. Lawrence Livermore National Laboratory is currently developing an x-ray microscope that uses high-efficiency reflective (Wölter Type I) optics for imaging millimeter-scale parts at resolutions of better than one micrometer. The optics use multilayer technology to increase the x-ray grazing angle, improving the efficiency of the optics.

The Wölter [1] imaging optic focuses x-rays that exit the sample (object plane) onto a scintillator (image plane). The scintillator converts the x-rays into visible light, which can be detected and imaged with a CCD camera. Our optic has a magnification of twelve. The distance between the sample and scintillator is five meters. Figure 1 shows the schematic of the microscope.

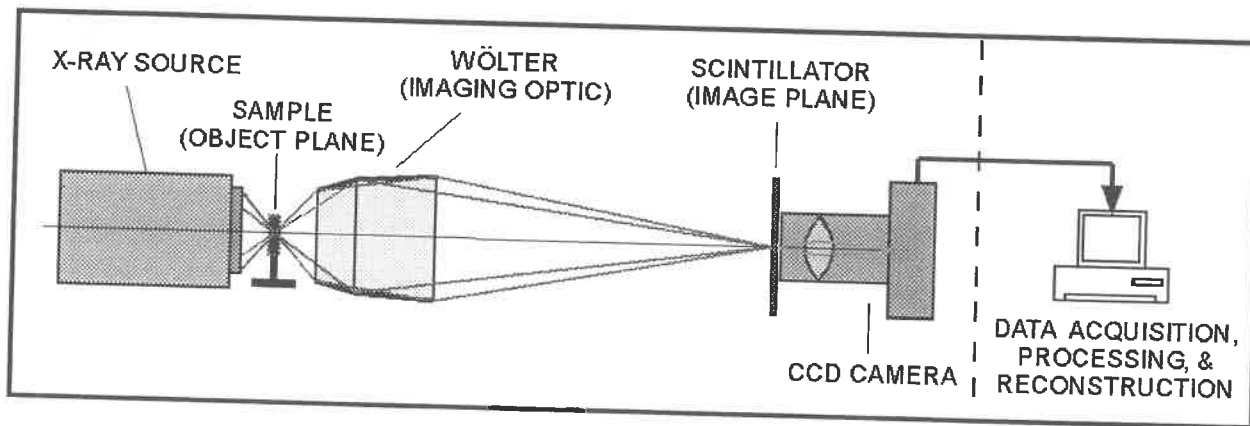


Figure 1: Schematic of our x-ray microscope

The Wölter optic consists of hyperbolic and elliptical reflective surfaces. This combination of reflective surfaces was first described by Wölter [1] in 1952. Although simple in concept, the fabrication of a high-quality imaging Wölter optic has proven to be difficult due to the tight tolerances on figure and roughness. In the remainder of this paper, we will describe the technique used to fabricate the optic and the challenges encountered.

2. Fabricating the Optic

To achieve the desired resolution, the optical design has very tight profile (figure) tolerances (~8 nanometers). Moreover, in order to achieve high efficiency, the surface roughness should be kept under 0.5 nanometers rms to minimize the scattering of x-rays. The optical surfaces on a Wölter optic are internal features. It is extremely difficult to machine and polish an external surface to our desired specifications; machining and polishing internal surfaces to our specifications is nearly impossible. Therefore, to meet imaging requirements, optical replication is being utilized.

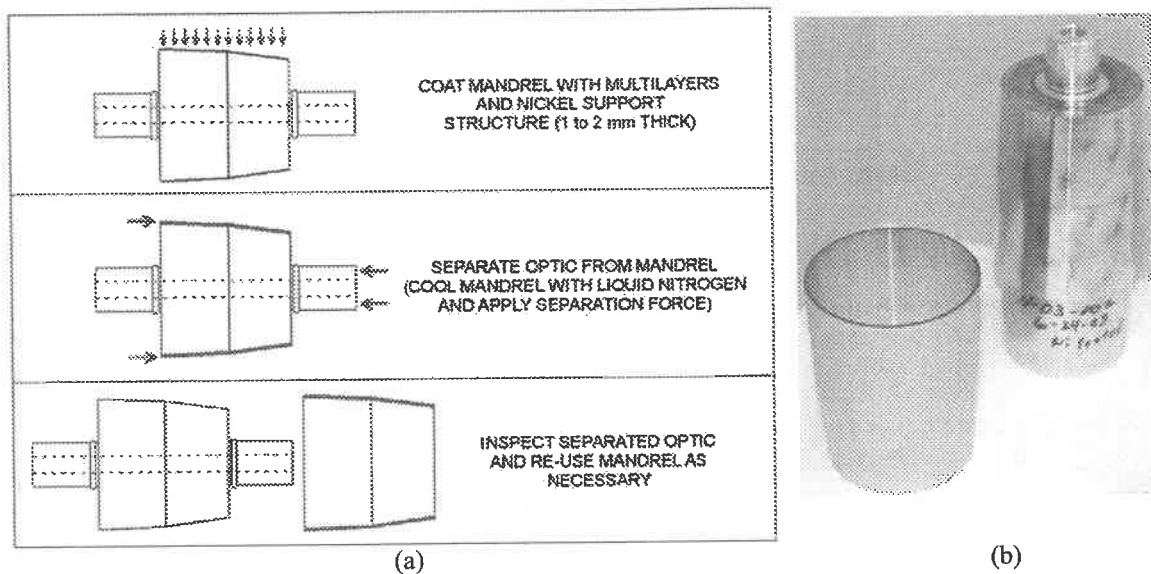


Figure 2: (a) replication procedure, (b) a replicated optic and its mandrel

The optical replication [2, 3, 4] method involves coating the outside surface of a high-accuracy super-polished mandrel and separating the coating (i.e. the optic) from the mandrel (see Figure 2). The coating consists of reflective multilayers and a 1.5-millimeter thick nickel substrate. The multilayers and nickel substrate are then separated from the mandrel. This process produces x-ray optics with similar figure and roughness properties to that of the high-accuracy mandrel.

X-ray optics are extremely sensitive to slope errors in the optical surface profile. Figure 3 shows the error created at the image plane by a slope error in one surface. The error, D , at the image plane is related to the slope error, α , at the mirror by the equation

$$D = L * 2 * \alpha$$

where L is the distance between the optical surface and the image plane. In our case, we do not want D to be greater than 6 micrometers; this corresponds to an error at the object plane of 0.5 micrometers (our microscope design has a magnification of 12x). The optical design process set L at 4583 millimeters. In our case we have two reflective surfaces, and each surface has an effect on D . Using an rms error analysis approach, we can combine the errors caused by both surfaces into a single equation:

$$D = \sqrt{(L * 2 * \alpha)^2 + (L * 2 * \alpha)^2} \text{ or } D = \sqrt{2} * (L * 2 * \alpha)$$

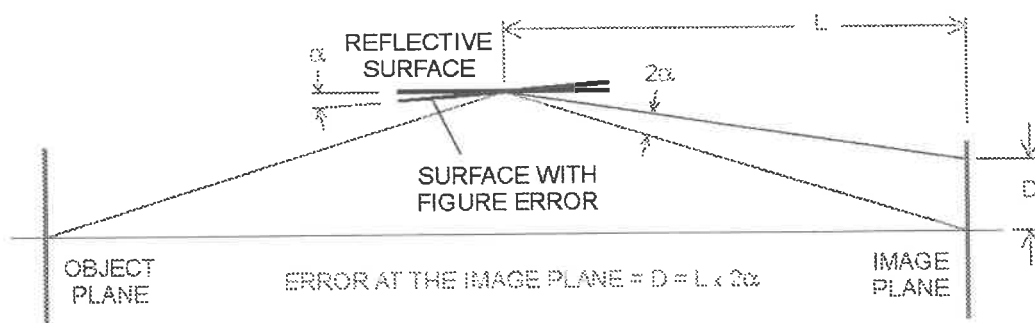


Figure 3: Relationship between figure error and image plane error

Rewriting the equation in terms of slope error, we get

$$\alpha = \frac{D}{2\sqrt{2} * L}$$

Using this equation, we get a slope error, α , due to figure errors only of 0.463 micro-radians or 0.10 arc-seconds. A 0.463 micro-radian slope error corresponds to an error in the surface profile of approximately eight nanometers. The eight-nanometer tolerance applies to higher order errors as shown in Figure 4; effects from size and linear slope errors over each surface are not as critical because these errors can nearly be corrected by moving the image plane towards or away from its designed position.

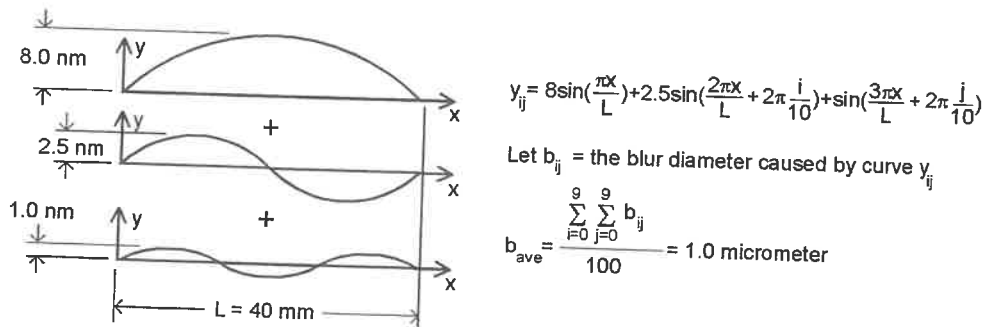


Figure 4: 8-nanometer errors in the profile can cause 1-micrometer imaging errors

3. Fabricating the Mandrel

Fabrication of the mandrel requires several steps: rough cutting the mandrel out of aluminum, nickel-plating the surface, diamond-turning the desired profile, and, finally, super-polishing the surface to a roughness under 0.5 nanometers rms. The diamond turning was done on LLNL's PERL II diamond turning machine (see Figure 5).

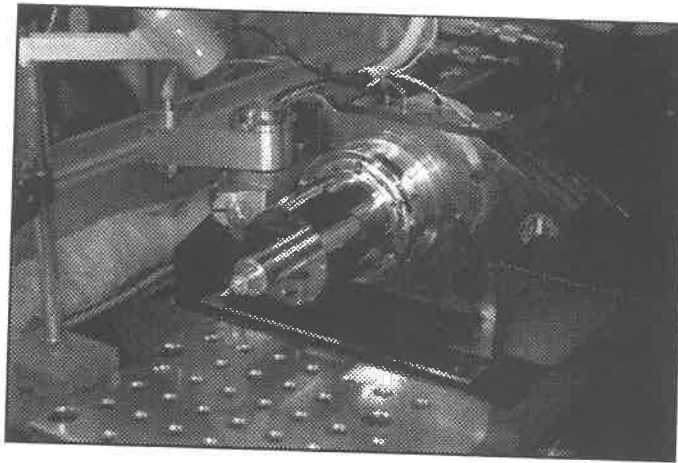


Figure 5: Diamond turning the mandrel

The size of the mandrel will be measured on the LODTM (Large Optic Diamond Turning Machine) in the future. The profile of each surface was measured with a Fizeau interferometer. The measured profiles deviated from the desired profiles by approximately 100 nanometers, which is an order of magnitude above what we want. We are currently diamond turning a second mandrel on the LODTM to improve the performance of this step.

Our first mandrel is currently being super-polished by an outside vendor. Getting the 0.5-nanometer roughness is going to take considerable effort. We currently are getting roughness measurements of between 0.6 and 1.5

nanometers rms. Moreover, some of the polishing compound is getting embedded in the surface, which could cause problems during the replication process. We are confident that these issues will be resolved in the near future. Figure 6 shows an Atomic Force Microscope (AFM) measurement of the hyperbola.

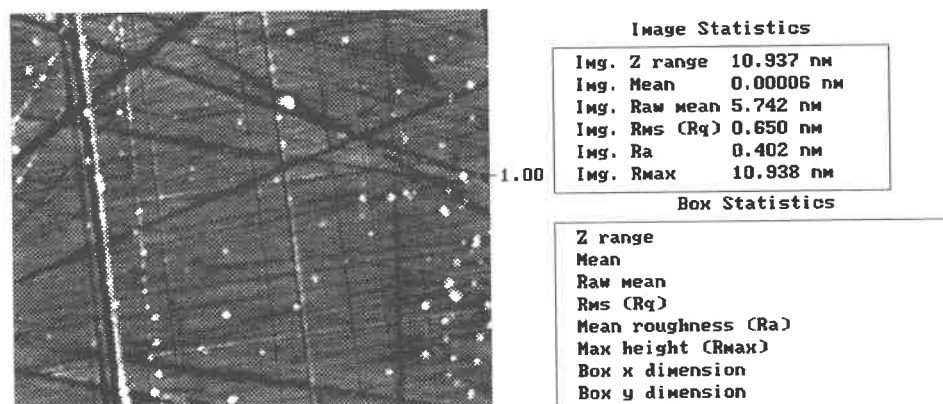


Figure 6: AFM roughness measurement of the hyperbola

4. Conclusion

Machining and polishing the mandrel to the desired tolerances is difficult. Also, limitations in techniques for measuring the mandrel dimensions are making it difficult to verify the results of the manufacturing processes. We will be investigating methods and tools for addressing our fabrication and metrology limitations. Regardless, we will be producing a replicated optic soon, and we will be building the prototype x-ray microscope in the spring of 2004 (see Figure 7). We will incorporate new optics into the microscope as the mandrel fabrication processes and optics improve.

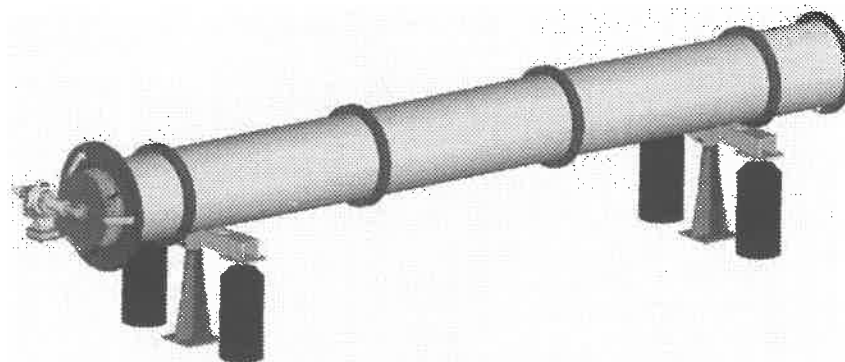


Figure 7: 3-D CAD model of the x-ray microscope

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Pinpoint chemical vapor deposition of carbon nanowire for nanometer-scale electronic devices

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1 Introduction

This paper reports the method to generate a carbon wire of tens nanometers in diameter (carbon nanowire) in a desired ("pinpoint") position for nanometer-scale electronic device. Recently, research which fabricates single-electron devices with carbon nanowire is widely done[1][2]. However, by the conventional methods, such carbon nanowire can not be generated in a desired position. So we developed pinpoint chemical vapor deposition (pinpoint CVD) technique using the electron beam of SEM(scanning electron microscope) to generate a carbon nanowire in a desired position under observation.

2 Concept of pinpoint CVD

It is known that when the electron beam of SEM is irradiated to an object, the residual-gas molecules near the object (mainly consist of carbon, hydrogen, oxygen) is excited and dissociated, and the dissociation is deposited on the object[3][4]. We applied this phenomenon to generation of the nanowire of free form in the air within a field perpendicular to the beam by scanning the electron beam from an object to the air slowly. Furthermore, 3-dimensional wiring is also possible by tilting a sample stage of SEM[5].

To apply this carbon nanowire to electron devices, we measured its electrical property. We generated a nanowire between two electrodes with 2 μm of gap to measure its volume resistivity (Fig. 3), and acquired the voltage-current curves that Fig. 4 shows. Although a change with the passage of time considered to be based on moisture is large, the volume resistivity of the nanowire could be estimated at the order of 100 $\Omega \cdot \text{m}$. The high resistivity of carbon nanowire generable now is considered to be because for it to be amorphous. TEM (transmission electron microscope) image of the carbon nanowire indicates that it is amorphous (Fig. 5).

In order to use for various nanometer-scale electronic devices, generation of the crystal carbon nanowire, such as SWCNT, multi wall CNT, and so on, is also required. These are generable by controlling generation conditions (atmosphere gas, a catalyst, decomposition energy). Energy of electron beam of SEM (about 30 keV) is already enough for decomposition however, and on the other hand, heating mechanism is required because a catalyst is not activated unless heated. Then, we add gas introduction equipment, catalyst substrate, and heating equipment to the SEM chamber to generate crystal carbon nanowires. We call such generation technique "pinpoint CVD".

3 Design of equipment for pinpoint CVD

For pinpoint CVD, mechanisms to provide (1)source of carbon, (2)energy to decompose the carbon compound, and (3)heat to activate a catalyst are required. As above-mentioned, the electron beam of SEM is used as the mechanism to provide energy for decomposition of carbon compound, and the other mechanisms are attached to the SEM chamber (Fig. 6). A nozzle (stainless steel pipe of 80 μm of bores) and a needle valve are prepared as a gas introduction mechanism, and ethanol gas is used as the source of carbon.

Since temperature enough to activate a catalyst is guessed more than about 600 degrees centigrade, it is important to protect SEM from thermal damage. The marginal temperature of objective lens of the SEM that we used (S-4000, by HITACHI Co., Ltd.) is 100 degrees centigrade, and the marginal temperature of the SEM stage is 60 degrees centigrade. Therefore, it is necessary to reduce the size of a heating part and

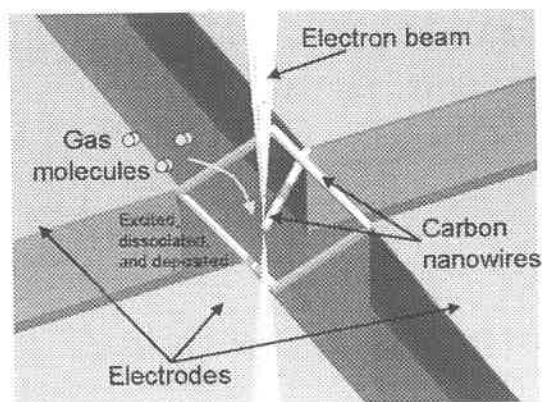


Fig. 1: Concept of pinpoint CVD

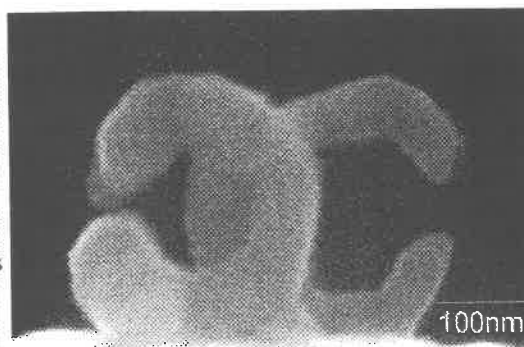


Fig. 2: Example of carbon nanowire

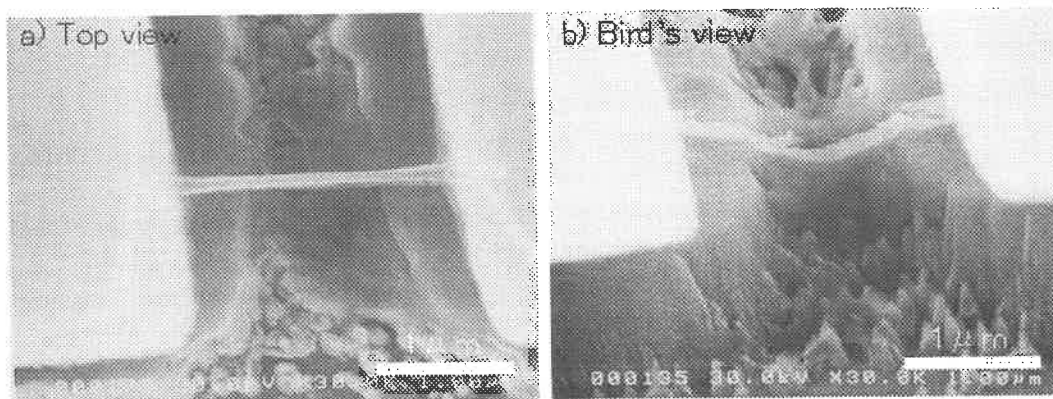


Fig. 3: SEM view of test chip for measurement of resistivity of carbon nanowire

the size of the contact section with the heating part as much as possible to decrease heat conduction, and reflecting plates are required to decrease thermal radiation. Then, the tungsten ribbon with the thickness of 25 micrometers, a width of 750 micrometers, and a length of 15mm was adopted as the heating part, and ceramic pipe($\phi 3\text{mm}$, $L=2\text{mm}$) and ceramic adhesives were used for fixation of the heating part. In order to interrupt radiant heat, the stainless steel pipe of 5mm of bores and the stainless steel pipe of 9mm of bores have been arranged around the heating part, and two stainless steel plates with a thickness of 0.5mm and 50mm square have been arranged to each upper and lower sides of the heating part as reflectors. Fig. 7 shows schematic of heating unit, and Fig. 6 shows photographs of it. We used a zeolite as a catalyst, and applied it directly on the surface of the ribbon heater.

4 Experiment

Since the marginal degree of vacuum of the specimen chamber of SEM is about $6 \times 10^{-2}\text{Pa}$, the needle valve of an ethanol gas introduction mechanism is adjusted so that the degree of vacuum of the chamber may be set to about $5.2 \times 10^{-2}\text{Pa}$. Next, the current of about 2 A is impressed to the heater so that the temperature of a heater might turn into 500 degrees centigrades. However, the picture of SEM became unstable and CVD was not able to be performed. The cause is considered because temperature control of a heater is imperfect, a part of heater was overheated, and the detector of SEM was saturated with the thermoelectron from there. Therefore, positive thermo control of the heater and protection of the detector

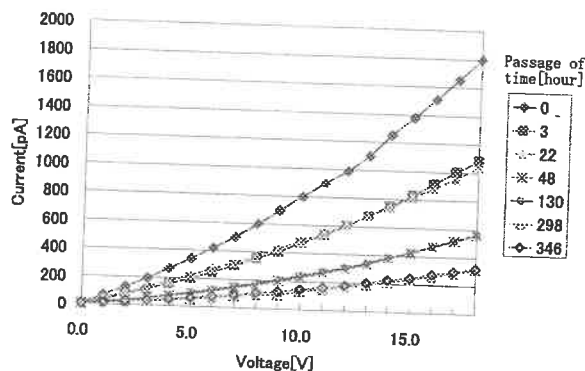


Fig. 4: Graph of voltage vs. current

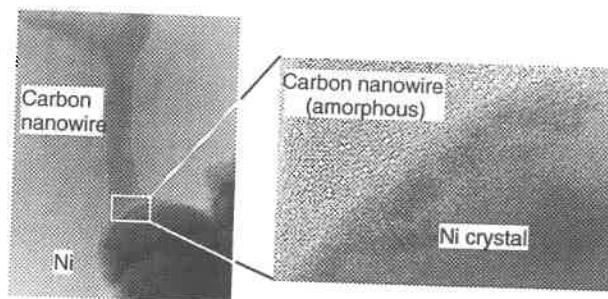


Fig. 5: TEM view of carbon nanowire on Ni substrate

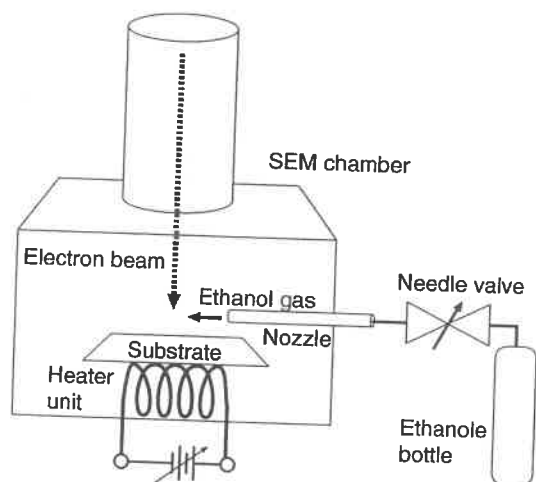


Fig. 6: Schematic of equipment for pinpoint CVD

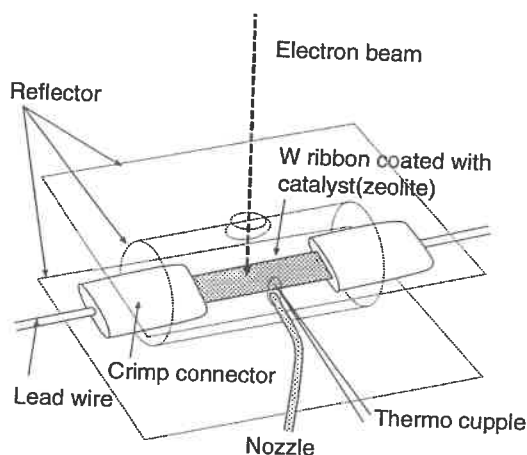


Fig. 7: Schematic of heating unit

are future subjects.

Next, we tried the experiment for applying amorphous carbon nanowire to an electron device. Because the amorphous carbon nanowire has large resistance compared with a single wall carbon nanotube(SWCNT) which conventionally used for research of single-electron devices or wiring, this nanowire is considered to be suitable for the device which operates by electric fields, such as field emission device. As an example, the nano vacuum tube with 500nm gapped electrodes was made (Fig. 9), and durability of the nanowire was evaluated. The breakdown test was performed having applied the voltage of about 50 V, and the result indicated that the nanowire has higher durability than an Au metal thin film of 50nm in thickness. We plan to measure the electric discharge efficiency of a nanowire and to control of the nano vacuum tube by electric field from now on.

5 Conclusion

We developed "pinpoint chemical vapor deposition" technique using the electron beam of SEM(scanning electron microscope) to generate a carbon nanowire in a desired position. Since the carbon nanowire gener-

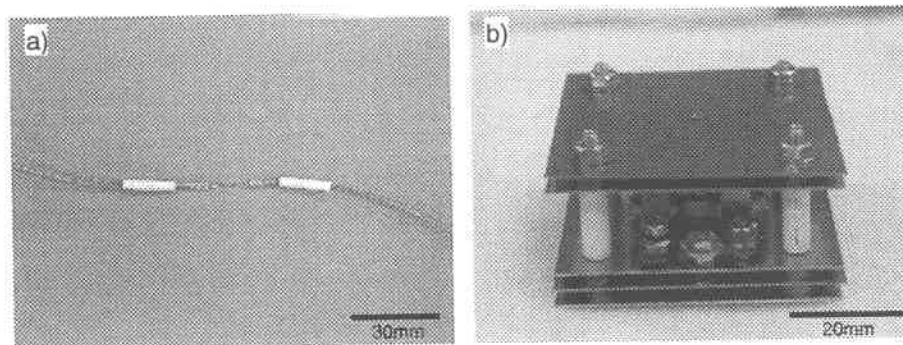


Fig. 8: Photographs of heater: a)Heating part, b)Reflectors

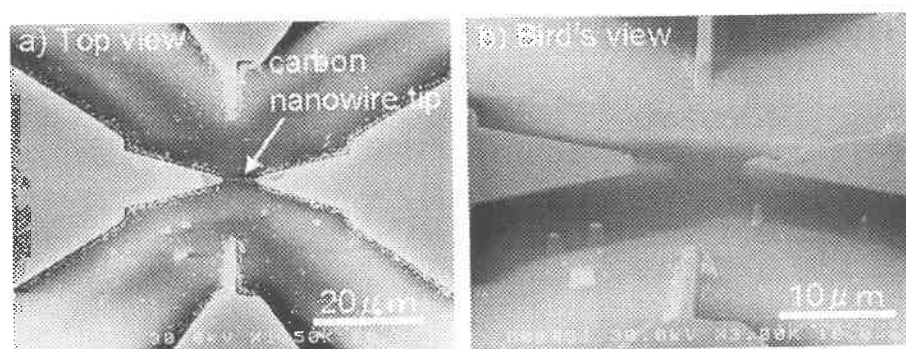


Fig. 9: Example of field emission device(SEM view)

ated using the residual gas in SEM as a source of carbon was amorphous, we made ethanol gas introduction equipment and catalyst heating equipment, and attached them in SEM. Although the experiment which generates a crystalline nanowire using the equipments was carried out, generation of a nanowire couldn't be performed because of the thermoelectron from a heater. Protection of the detector of SEM from the thermoelectron and positive temperature control are performed, and the conditions which generate a crystalline nanowire will be investigated from now on.

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Manufacturing and Quality Control of the NIST Reference Material 8240

Standard Bullet

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EXTENDED ABSTRACT:

The National Integrated Ballistics Information Network (NIBIN) is currently under development by the Bureau of Alcohol, Tobacco and Firearms (ATF) and the Federal Bureau of Investigation (FBI). In support of this effort, the National Institute of Standards and Technology (NIST) is developing both physical and virtual reference standards. These are designed to be used by crime laboratories to verify that results obtained when using the NIBIN meet legal requirements and that the NIBIN equipment is operating properly. It is also intended to use these standards for the establishment of ballistics measurement traceability to NIST, ATF and FBI [1].

One of these reference standards is the NIST RM (Reference Material) 8240 standard bullet shown in Fig 1. The surface profile of six master bullets from ATF and FBI were measured at NIST using a stylus measurement system. The profiles have a 0.25 μm point spacing. The resulting set of six digitized 2D profile signatures are used as a 2D virtual bullet signature standard for the production of standard bullets using a numerically controlled diamond turning machine at NIST. These standard bullets are designed to have nearly identical 2D signatures, and will be used for instrument calibration and measurement quality control for the NIBIN. This document focuses on the manufacturing and quality control of these standard bullets.

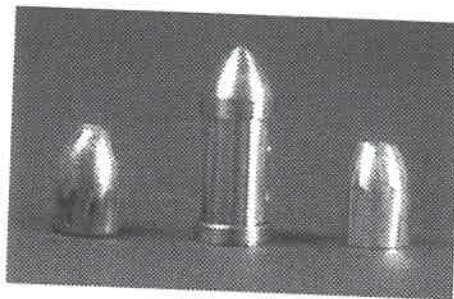


Figure 1: A master bullet from ATF National Laboratory Center (left), a NIST prototype standard bullet (center), and a NIST RM 8240 standard bullet (right).

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Commercial equipment and materials are identified in order to adequately specify certain procedures. In no case does such identification imply recommendation or endorsement by NIST, nor does it imply that the materials or equipment identified are necessarily the best available for the purpose.

Because the bullets are manufactured by diamond turning the profile signatures onto the bullet lands, bright acid copper is the preferred material for electroforming the substrate. The deposit is electroplated onto copper cylinders as shown in Fig 2. The electrolyte is a basic copper sulfate electrolyte, to which proprietary agents have been added. These agents serve not only as grain refiners, but also level the deposit on a microscopic scale. The deposit, shown in Fig 3, is specular and ductile as plated, with a sub-micron, equiaxed, crystalline grain structure. The bright acid copper electrolyte has a high deposition rate of 1.25 $\mu\text{m}/\text{minute}$.



Figure 2: OFHC copper blanks, separated by spacers, destined to be electroplated and then machined into reference bullets.

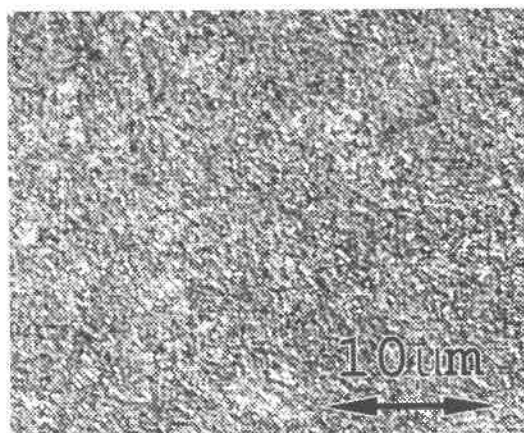


Figure 3: Photomicrograph of electroplated surface showing sub-micron grain structure.

Once electroplated, each bullet is mounted on a small cylindrical arbor and turned on a conventional lathe to near net shape. Next, up to 20 bullets are mounted to the face plate of the diamond turning machine spindle as shown in Fig 4. One of the profile signatures is machined on to the bullets. They are then rotated and remounted, and the next profile is machined. This process is repeated for all six profiles. The photomicrograph in Fig 5 compares a portion of a land from 2 different bullets. It shows that, qualitatively at least, there is a very good match in the land patterns.

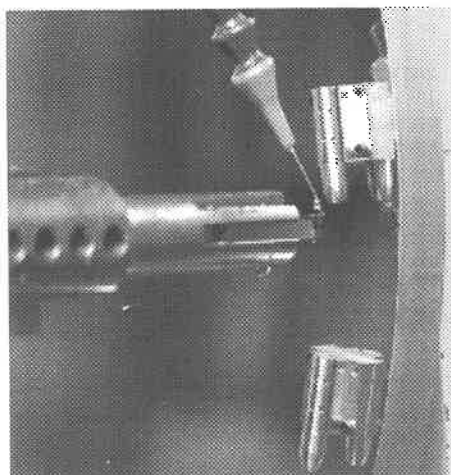


Figure 4: Machining lands using diamond turning machine.

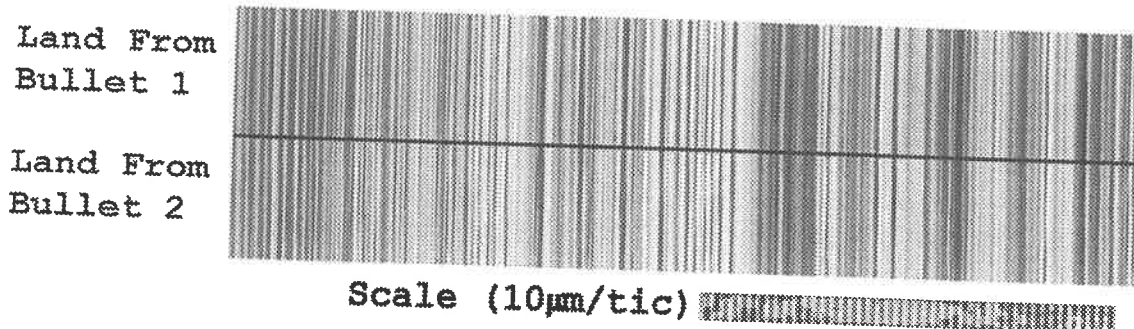


Figure 5: Photomicrograph of a portion of the land area from two different reference bullets shows excellent reproducibility.

Next is the measurement and quality control phase of the production process. In order to verify the quality of the bullets, each is measured using a stylus measurement system. The desired profiles (the 2D virtual bullet signature standards noted above) and the measured profiles are compared using a NIST developed parameter based on auto- and cross-correlation functions [2]. A minimum of 95% correlation is required, but often the correlation is better than 99%. There are two paths of data flow through the quality control process as shown in Figure 6 below. The left-most path shows the data flow of the virtual signature standard, while the right path shows the data flow of the signature of the manufactured bullet being checked.

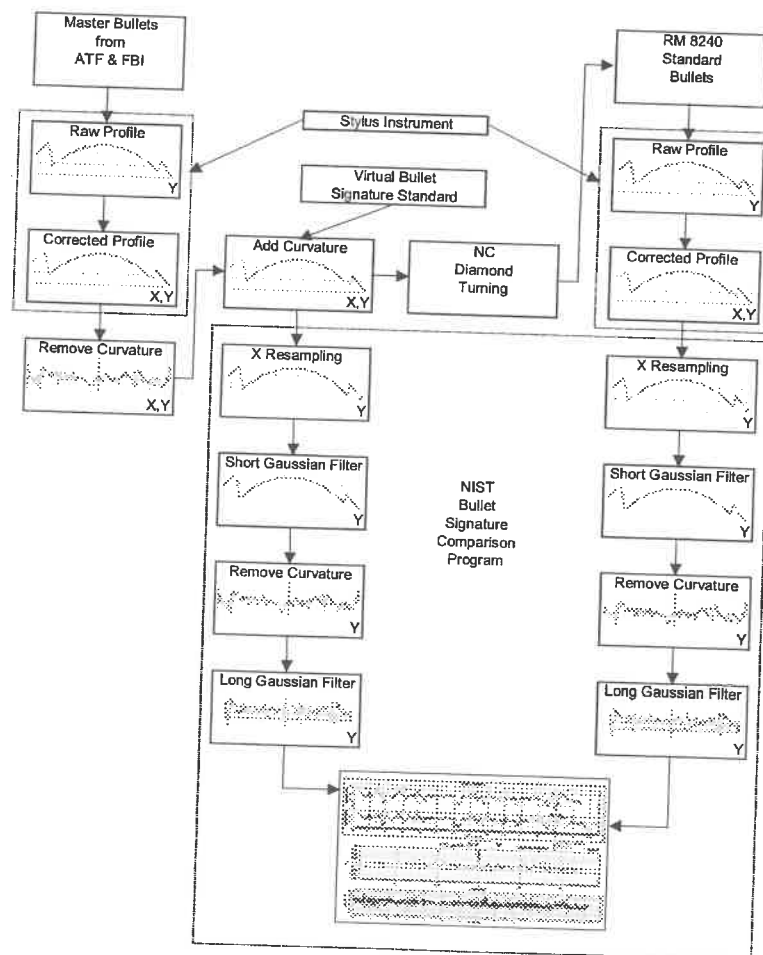


Figure 6: Flow of data in the quality control process.

Both paths emanate from the six different master bullets from the ATF and the FBI. As a result, the radius of curvature for each of the lands is initially different. The time required to machine the lands using the diamond turning machine is minimized when the radius of those lands are the same as the radius of the standard bullet itself. In order to minimize the machining time, the radius was removed from each of the six lands and the radius of the standard bullet added back to each land. By doing this, the virtual bullet signature standard is generated.

As the bullet signatures were measured on a highly curved bullet surface, the total range of Z heights in the profiles is significant. The stylus instrument initially acquires the profiles as points equally spaced in the X direction. However, as with most stylus instruments, the stylus actually measures the angle of the stylus tip as it rotates about a pivot point and not the Z height. When the stylus instrument converts these angles into true Z heights, the once equally spaced points become unequally spaced. Unequally spaced points are not a problem when manufacturing the bullets, but most profile analysis techniques are significantly easier to implement with equally spaced points. Fortunately, cubic spline functions may be used to resample the profile data to make the profiles equally spaced again. Cubic splines were chosen over linear interpolation because linear interpolation yields results with a systematic bias towards being too small.

Before computing the correlation between the signatures, the virtual bullet signature and the signature of the bullet being tested are both filtered so that the features with wavelengths either too small or too large to be important are removed. To accomplish this, first a low-pass Gaussian filter with a 0.0025 mm cutoff length is used to remove high frequency noise. The large scale curvature is removed by fitting and then subtracting a circle. Finally, a 0.25 mm long cutoff low-pass Gaussian filter is used.

In conclusion, both the visible comparison and the correlation measurements show that the profiles diamond turned on to the standard bullets reproduce well and can be produced in relatively large quantities. This allows different forensic laboratories to use a common physical artifact. It should be noted that this research involved samples a few months old. Future research will involve checking the long term stability of the profiles.

This work was supported in part by the National Institute of Justice through the NIST Office of Law Enforcement Standards.

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Key words: reference material, ballistics, diamond turning, electroplated, correlation function, reference bullets

ELECTROMAGNETICALLY DRIVEN FAST TOOL SERVO

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INTRODUCTION

Fast Tool Servo (FTS) technology can enable precisely manufacturing complicated surfaces with nanometer-scale resolution requirement. Such surfaces are used in a wide range of products, including films for brightness enhancement and controlled reflectivity, as well as in micro-optical devices such as Fresnel lenses and microlens arrays. The limits on stroke, bandwidth, acceleration, and position noise of the FTS impose limits on the types, quality, and rate at which the intended surfaces can be produced. The requirements for high throughput drive simultaneously the need for high bandwidth, high acceleration, and accuracy for the FTS.

Most of the high bandwidth short stroke FTS's are based on piezoelectric actuators [1][2][3]. Piezoelectric FTS's have the advantage that they can readily achieve bandwidths on the order of several kHz and high acceleration on the order of hundreds of G's, are capable of nanometer resolution of positioning, and can achieve high stiffness (usually greater than $50\text{N}/\mu\text{m}$ in the typical sizes used). However, piezoelectrically actuated FTS's also have significant disadvantages. When the piezo materials undergo deformation, heat will be generated by the hysteresis loss, especially in high bandwidth and high acceleration applications. Another problem is that the structural resonance modes of the PZT stacks limit the working frequency range, because operation near internal resonances will cause local tensile failure of the PZT ceramics.

However, electromagnetic actuators do not have such problems and thus are a promising alternative. Electromagnetic force density can be as high as $9 \times 10^5 \text{ N/m}^2$ at 1.5 Tesla flux density, and this can possibly generate a 4000G acceleration on a 3mm thick iron disk. This research focuses on developing an electromagnetically driven FTS as a replacement for the widely used piezoelectrically actuated FTS's. The experimental results of the first prototype show that the electromagnetically driven FTS is very promising for high bandwidth application.

DESCRIPTION OF FTS DESIGN

Figure 1 shows the configuration of our electromagnetically driven fast tool servo. The backbone of the moving assembly is a carbon fiber tube. On its rear end is attached a disk-shaped armature and the cutting tool tip is installed on the front end. The whole moving assembly is suspended to the FTS frame by two flexures made from spring steel sheet. As shown in Figure 2, these flexures guide the tool tip moving along only one degree of freedom in the tube axial direction with the other five degrees of freedom constrained. The armature is push-pull driven by a pair of circular E-type solenoids in the rear and front sides. The air gaps between the armature and solenoid surfaces are set at $100\mu\text{m}$ to ensure easy fabrication, though a smaller air gap will improve the energy efficiency. The coil windings are implanted into the slots of the solenoids and these slots are 2mm wide and 20mm deep to ensure enough Ampere-turns. The coils are made from 4 strands of gauge #30 self-bonding wires to reduce the skin-depth effect in the copper conductor at high frequency operation. A $\pm 100\text{V}$ 2A 100kHz bandwidth linear power amplifier is designed to drive the coils. A 100kHz bandwidth capacitance probe is installed in the rear side of the FTS to sense the motion of the armature. In this design, in order to reduce the mass of the moving assembly and thus reduce the reaction force, the size of armature is minimized and carbon fiber is

used as the backbone. To reduce the eddy current induced along the magnetic flux path, the materials for both the armature and the solenoids are sintered soft magnetic material made from iron particles of about 100 μm in diameter.

DESCRIPTION OF CONTROLLER DESIGN

The electromagnetically driven actuator is difficult to control in the sense that the actuating force is proportional to the current squared and inversely proportional to the air gap squared. Moreover, the force will decrease with frequency because the magnetic field cannot penetrate the magnetic material at high frequencies. In order to compensate these nonlinear and frequency dependent characteristics, a dynamic nonlinear compensation (DNC) method shown in Figure 3 is applied. Here $K(x)$ represents the relation between current and magnetic field, $D(s)$ the eddy current effect, and the "Square" block relates the magnetic flux to the actuating force. This DNC is used to partially compensate the nonlinearity of the actuator, but is not expected to linearize the actuator completely because it is a feed-forward model-based method and modeling errors always exist. The whole position control loop is compensated with a lead-lag controller and low-frequency

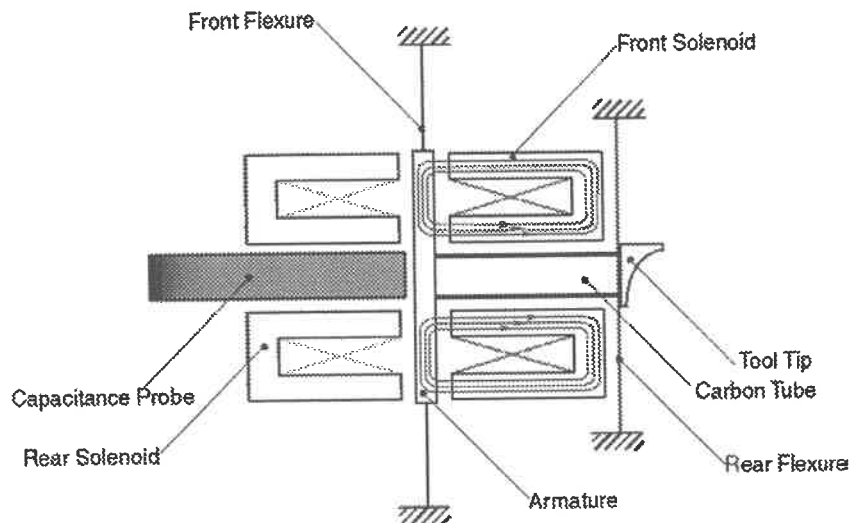


Figure1: Configuration for the electromagnetically driven FTS

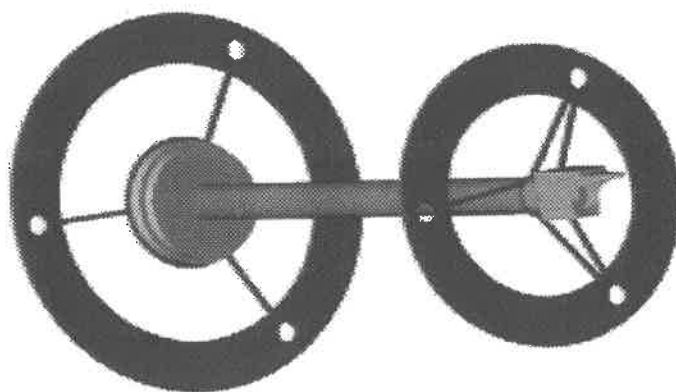


Figure 2: Flexures for the moving assembly

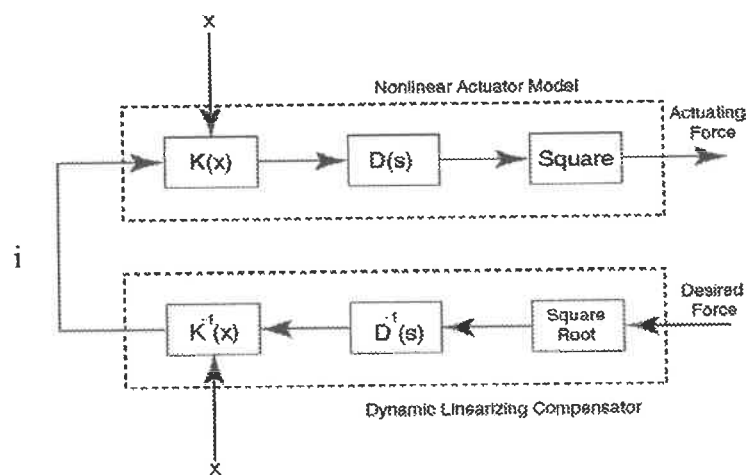


Figure 3: Dynamic Nonlinear Compensation

integrator. At the spindle rotational frequency and its higher harmonics, a plug-in type adaptive-feedforward-compensation (AFC) controller is used to improve the rejection of spindle-generated disturbance and to improve the spindle-synchronized trajectory tracking performance. The whole controller is shown in Figure 4.

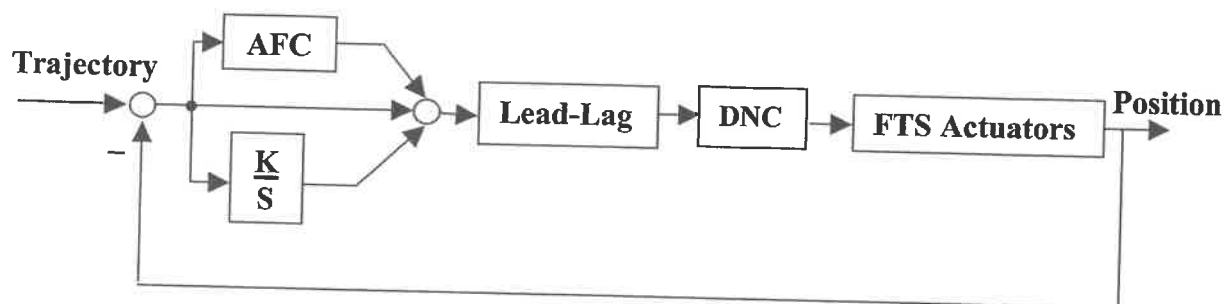


Figure 4: Controller Structure

EXPERIMENT AND RESULTS

The controller is implemented with a DSPACE 1103 board, and all the digital controllers are in the discrete domain. The full stroke of 50 μm can be achieved up to 1kHz operation. The maximum acceleration is 160 G's when tracking a 9 μm peak-to-valley 3kHz sine wave. For a sampling frequency of 100kHz, the closed-loop frequency response is shown in Figure 5 and the small signal bandwidth can be as high as 10kHz. For a sampling frequency of 83kHz, the closed loop bandwidth is 8kHz. The 100nm closed-loop step response is shown in Figure 6.

The position error is 1.2nm RMS when the spindle is turned off. After the spindle is turned on, the error degrades to 3.5nm RMS because of the PWM noise from the spindle amplifier. To evaluate the tracking performance, a 10- μm peak-to-valley 1 kHz sine wave trajectory is applied to drive the fast tool servo. When the AFC is not plugged into the control loop, the tracking error is 1.048 μm RMS. When the first harmonic AFC is applied, the error is 0.0214 μm RMS. The tracking error reduces to 0.0148 μm RMS when the second harmonic AFC is further applied and to 0.0073 μm RMS when the third harmonic is also added. This shows that the nonlinearity of the actuator and the power amplifier will introduce disturbance

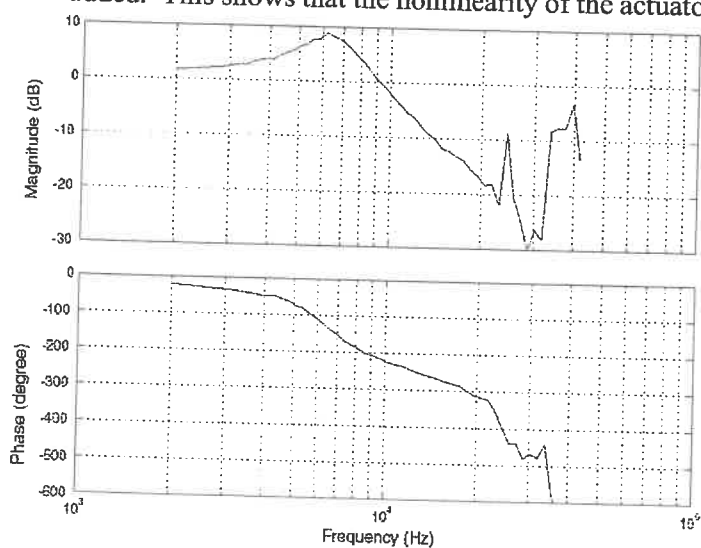


Figure 5: Closed loop Frequency Response

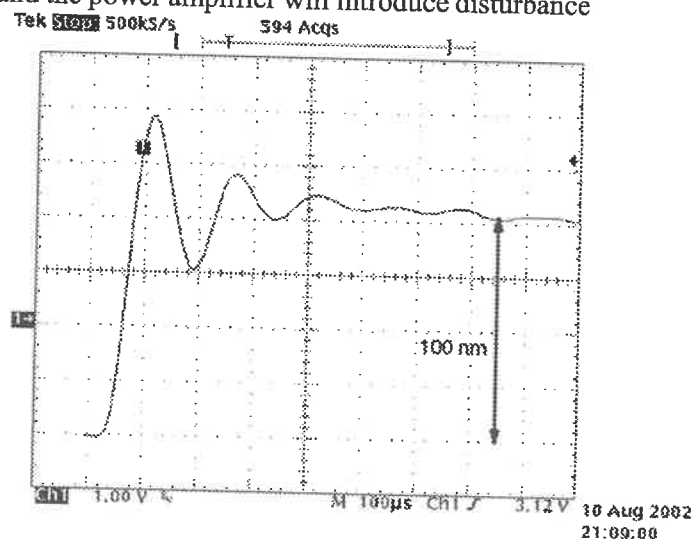


Figure 6: Small signal step response

forces of second and higher order harmonics, and the AFC of poles at multiple harmonic frequencies can significantly improve the tracking error.

A cutting experiment is conducted on a Moore diamond turning machine. The same DSPACE 1103 board controls both the X-Z slides of the machine and the FTS. A multiple sampling rate system is implemented. The sampling rate for the FTS controller is 83kHz and the sampling rate for the spindle and X-Z slides controller is 4kHz to ensure that the slides controls achieve 100Hz bandwidth. As shown in Figure 7, the sinusoidal surface of 30 harmonics per revolution is faced on a piece of aluminum material.

In conclusion, the electromagnetically driven fast tool servo is promising to achieve high bandwidth and high acceleration performance for complicated surface manufacturing.

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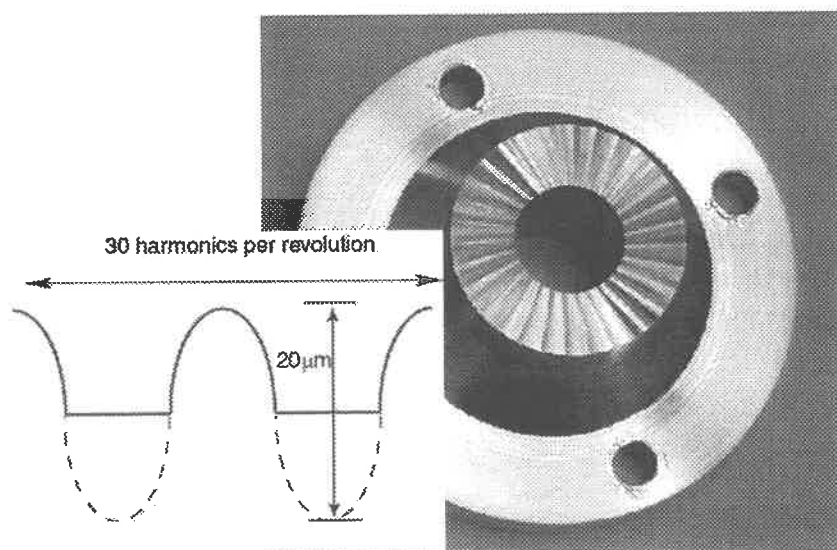


Figure 7: Diamond turned part by the electromagnetically driven prototype FTS. The surface is machined by face turning. The spindle speed is 1800rpm. The profile expanded in the circumference is half sinusoidal wave, as illustrated in the bottom left curve. There are 30 harmonics for each spindle revolution, and the peak to valley amplitude of the sine wave is 20 μm . The flat surface was machined first and then the sinusoidal surface was cut.

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Error Compensation Using Inverse Actuator Dynamics

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A signal processing technique has been developed to dramatically increase both the usable bandwidth and tracking accuracy of an actuator. Traditional feedback controller design makes little or no use of the information content of the driving command signal. Feedforward and look-ahead schemes use a small sliding window of trajectory data to improve tracking performance and to reduce the overshoot of an accelerated axis. If the entire command signal for each axis of a machine is generated off-line, and the dynamic response of each axis is well characterized, then a new command signal can be derived which will drive the axes to follow the correct motion path. This idea is related to reference feedforward control, but by considering the complete command signal *a priori*, performance of the overall machine tool is greatly enhanced.

An inverse dynamics algorithm has been developed that modifies a motion path using an actuator's impulse response to produce a new command signal that counteracts actuator dynamics, yielding a significant reduction in steady-state tracking error. This technique critically depends on knowledge of the actuator dynamics and does not replace the feedback controller. Rather it improves the response of an actuator by compensating for high frequency dynamics that the servo feedback loop cannot correct due to saturation of the drive system.

Background A diamond turning lathe is capable of producing high quality surfaces in a variety of nonferrous materials. However, its use is normally limited to surfaces of revolution that are realized by moving the tool and/or workpiece through one-half of a cross section of the desired shape while the rotation of the workpiece on a spindle causes a symmetric surface to be machined. Advantages of turning are reduced process times as compared to milling due to the continuous high cutting speed achieved via rotation of the workpiece, excellent form fidelity and optical quality surface finish.

It is desirable to produce optical systems with the diamond turning process that have high spatial frequency features or are non-rotationally symmetric (NRS). The addition of a high-bandwidth fast tool servo (FTS) is one technique that has been used to add low amplitude, sub-revolution features to an asphere or sphere. Examples include torics and segmented off-axis conics. Two major complications arise when using a FTS to produce NRS surfaces: static decomposition and dynamic convolution.

Static Decomposition First, the motion commands for each axis of the base machine and the FTS must be generated from a 3D description of the desired part. In some cases the location of the part with respect to the spindle axis and the orientation of the FTS are also considered. The part shape must be decomposed into a rotationally symmetric surface of revolution (i.e., a cross section) for the diamond turning machine (DTM) and a spindle angle dependent motion path for the FTS. The principle difficulty is that while the FTS has a high relative bandwidth it has a limited range of motion. This decomposition process has been solved for a variety of shapes and will not be addressed in this paper.

Dynamic Convolution The second complication is the dynamic behavior of the axes, particularly the FTS. The heterogeneous nature of the axes results in phase errors in the recomposition of the decomposed command signals. The phase errors of the base machine axes can usually be ignored as the axis accelerations are moderate and velocity tracking performance of the control system is quite good. However even a small lag time associated with a FTS will result in poor form fidelity for off-axis surfaces and improper placement of NRS surface features with respect to the rotationally symmetric (RS) base surface and any fiducial. The decomposed FTS motion changes direction at least once on each rotation of the spindle whereas the axis motions are almost always in one direction (the waxicon being a notable exception). A first order correction of the FTS phase error is to apply a constant lead time to its motion commands. The magnitude of this phase advance depends only on actuator bandwidth and spindle velocity. However for motion paths that use a substantial portion of the bandwidth range or contain frequency components close to the bandwidth of the axes this simple technique yields poor results.

The response of a dynamic system to an applied signal is a function of the frequency of that signal. In general, the response results in an attenuated and delayed motion of the output with respect to the input signal. When the system is a fast tool servo and the dynamic input signal is made up of a variety of frequencies, the result is form error in the machined surface. The actuator mechanism can be considered as a spring-mass-damper driven by a periodic function $x(t)$ with an amplitude A and a frequency of ω radians per second. The steady-state output motion $y(t)$ is

Input:	$x(t) = \sum A_i \sin(\omega_i t)$
Output:	$y(t) = \sum a_i A_i \sin(\omega_i t + \phi_i)$
Modified Input:	$x(t) = \sum \frac{A_i}{a_i} \sin(\omega_i t - \phi_i)$
Modified Output:	$y(t) = \sum a_i \frac{A_i}{a_i} \sin(\omega_i t + \phi_i - \phi_i)$

amplitude attenuation and phase related delay must be eliminated. To achieve this, the input signals are modified prior to applying them to the system such that the attenuation is canceled and the phase is compensated.

After the dynamics of the system are identified and the desired motion is decomposed into sinusoids, amplitude (a_i) and phase (ϕ) adjustments can be found to construct a modified command signal that moves the actuator in the desired manner. A technique for system identification and an input signal modification algorithm have been developed and applied to the Variform™ FTS to produce the desired output response for high frequency inputs.

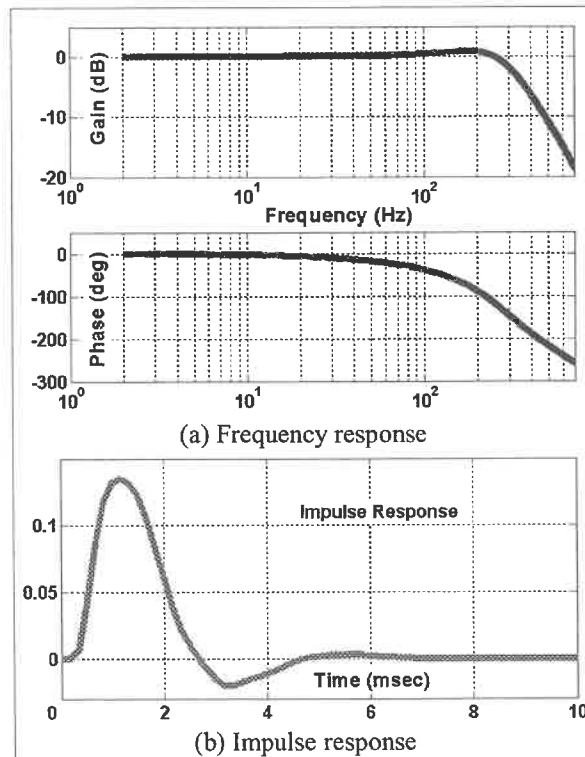


Figure 1. Dynamic response of the Variform

response $h[n]$ of the system. The convolution operation describes how a linear filter (an actuator) modifies a given input signal (command) to produce an output signal (motion). This process is represented by the * (star) operator and is expressed in Equation (1). Each value in the convolved output signal can be calculated with the summation in Equation (2), where M is the length of the impulse response. The length of $y[n]$ is $N+M-1$. Convolution is mathematically equivalent to polynomial multiplication and can be directly calculated from Equation (2) in the time domain. If the discretely sampled input and impulse signals are transformed to the frequency domain, convolution becomes a straightforward complex multiplication (element by element) of two vectors. The discrete Fourier transform (DFT) is commonly implemented by the fast Fourier transform (FFT) algorithm which exploits the symmetries in Equation (2) to accelerate this calculation. Equation (2) can thus be restated in the frequency domain using an FFT operator as shown in Equation (3). The inverse FFT operation can then be used to find $y[n]$.

attenuated by a gain factor a and is phase shifted by an angle ϕ .

If the input $x(t)$ contains multiple frequencies, the corresponding output can be determined using *superposition*; that is, the input signal can be decomposed into single-frequency components each of which will be attenuated and delayed. The response of the system $y(t)$ will be the sum of these frequency components. To machine a desired surface profile with high fidelity, the

Actuator Dynamic Response The dynamic response of the Variform can be presented in either the time domain (impulse response) or the frequency domain (frequency response). They are both shown in Figure 1 and are mathematically equivalent and complete for a linear system. On the left in Figure 1(a), the top graph shows the ratio of the amplitude of the output to the input (in dB units) as a function of frequency. At low frequency, the output is equal to the input and the ratio is unity giving a dB value of zero. As the frequency is increased, the amplitude response peaks at about 200 Hz and then drops rapidly for higher frequencies. The lower graph in Figure 1(a) shows the phase angle between the input and the output. At lower frequencies these signals are nominally in phase, but even a small lag of less than a degree represents a significant length of arc if the actuator is at a large radius from the axis of spindle rotation. As the frequency rises the output increasingly lags the input. At 100 Hz this lag is about 45° and at 340 Hz it is about 180°. The impulse response is shown on in Figure 1(b) and describes the characteristics of the system with respect to time. The Fourier transform of the impulse response is exactly equal to the frequency response and the inverse Fourier transform of the frequency response is the impulse response.

Deconvolution The effect of a dynamic system on an input signal of length N can be determined by applying the convolution operation to the input $x[n]$ and the impulse

$$x[n] * h[n] = y[n] \quad (1)$$

$$y[k] = \sum_{m=0}^{M-1} h[m] x[k-m] \quad (2)$$

$$\text{FFT}(x[n]) \times \text{FFT}(h[n]) = \text{FFT}(y[n]) \quad (3)$$

$$\text{FFT}(x[n]) = \frac{\text{FFT}(y[n])}{\text{FFT}(h[n])} \quad (4)$$

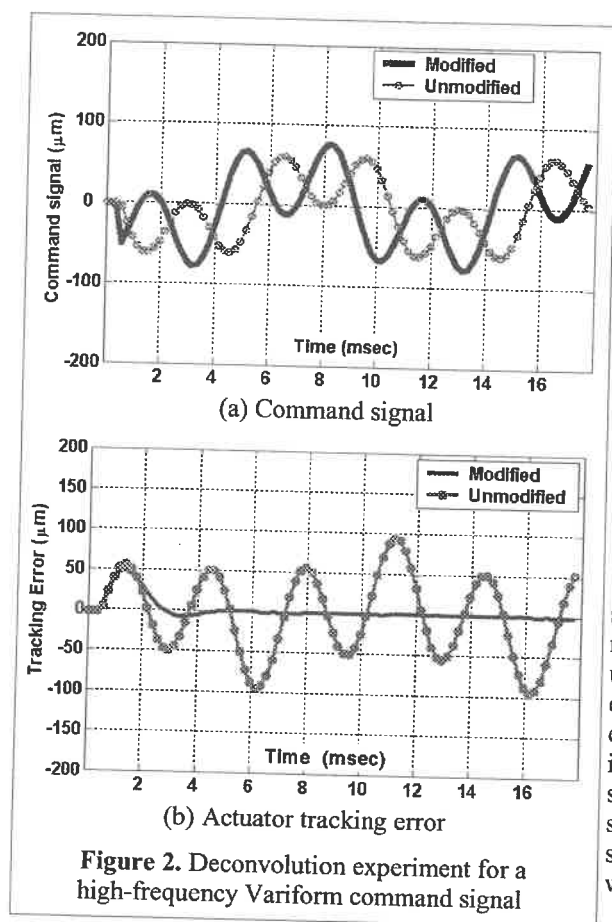


Figure 2. Deconvolution experiment for a high-frequency Variform command signal

Deconvolution is the inverse of convolution and can thus be used to find $x[n]$, given an actuator impulse response $h[n]$ and a desired motion $y_d[n]$. Rearranging Equation (3) gives Equation (4). Application of the inverse FFT operation then yields a signal that should exactly counteract the actuator dynamics so that it follows the desired trajectory, eliminating steady-state following error. This technique critically depends on knowledge of the actuator dynamics as embodied in the impulse response and requires the complete desired motion profile.

Experimental Verification A Variform FTS with 400 μm range and 350 Hz bandwidth has been used to perform experiments to validate the algorithm. The response of this actuator is predominately second order although nonlinearities are present for large amplitude command signals. Figure 2 shows the path error associated with a 160 μm command signal that is the sum of 100 Hz and 300 Hz sine waves. The response of the actuator to the 300 Hz sinusoid is 140 degrees out of phase and attenuated 2.5 dB resulting in a very large (200 μm PV) error. The algorithm uses deconvolution to calculate a new command signal that effectively precompensates for these effects individually at each constituent frequency. The following (or tracking) error is reduced by about 3 orders of magnitude. The response shows virtually no delay from the desired excursion after a startup interval lasting approximately 4 ms. In practice, this startup period will occur before machining begins and thus will not affect the fidelity of a surface.

Saturation In general, the dynamic response of a linear actuator is a function of the frequencies of the applied signals. Though the dynamic response results in an attenuated and delayed motion, the output of a linear system contains only the frequencies of the applied signals. Since the Variform FTS contains hardware limitations in which the velocity is constrained, the output profile is reshaped. The effects of this distortion on the operating range of the Variform as well as the measured impulse response employed in the deconvolution technique can be thoroughly investigated using frequency spectrum analysis. At frequencies near actuator bandwidth even a saturated input signal will not produce a full magnitude excursion. This situation is entirely predictable for a given command signal and is easily incorporated into the algorithm.

In Figure 3, the deformed shape of the FTS response to a high frequency input is illustrated. The distorted response comprises not only the frequency components of the desired trajectory (200 Hz), but also the 3rd and the 5th harmonics, 600 and 1000 Hz, respectively. The constituent sinusoid associated with the frequency of the desired path is in-phase and the 3rd harmonic is out-of-phase, transforming the output to a triangle-like profile. Note that the amplitude of this constituent sinusoid is smaller than the amplitude of the distorted output excursion.

The amplitude difference has a direct effect on the accuracy of the impulse response measurement using a spectrum analyzer. The analyzer generates a varying frequency input (i.e., swept sine wave) with fixed amplitude to the FTS, and analyzes the attenuation and phase of the actuator motion at each input frequency as measured by an LVDT. If the output signal is velocity saturated, its frequency component associated with the input frequency has smaller amplitude than the actual output signal. As a result, the frequency response of the Variform shows severe attenuation. Figure 4 depicts the attenuation of the Variform as a function of frequency and amplitude. The dynamic response shows significant attenuation where the input signal contains large amplitudes with high

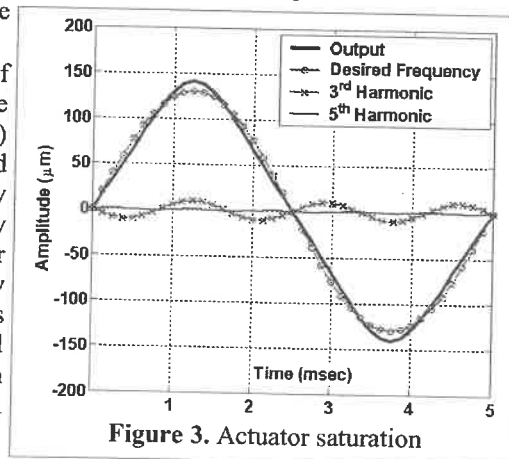


Figure 3. Actuator saturation

frequencies. This is because the maximum velocity of this FTS is limited to 140 mm/sec.

Real Time Deconvolution The desired path of the FTS can be defined as a function of the spindle speed and the axis cross-feed rate of the DTM. As either velocity changes, the time period of the desired path is also altered. Since deconvolution over the entire command signal is only valid for immutable machining parameters, a more robust approach would be to perform the

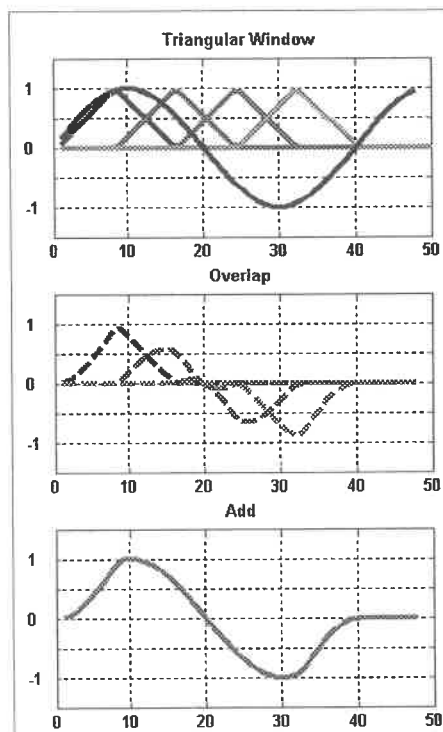


Figure 5. Overlap-add reconstruction

deconvolution in *real time* over a moving window into the desired trajectory. As shown in Figure 1b, the impulse response of the FTS is essentially zero after 10 ms. The modified actuator command signal at any given instant depends only on a window of the command signal that extends forwards and backwards in time over a 20 ms interval. A more advanced deconvolution algorithm has been developed to perform the deconvolution in *real time*; albeit at a sub-servo rate.

A long tool path must be divided into short pieces that are individually modified using the deconvolution technique. While the Variform follows the current input command, the next piece (or window) of the desired path is modified using the most recent parameters. This *real time* deconvolution can be achieved by applying a window function¹ to a long signal. In general, the window is short in length for sensitivity to high frequency signals.

Real time deconvolution transforms these short windowed pieces to the frequency domain where the operations of inverse dynamics occur. The transform using window functions is called the short-time Fourier transform (STFT) [1]. The reverse process, an overlap-add method is used to reconnect these short pieces after deconvolution. Reconstruction of short overlapped adjacent windows is illustrated in Figure 5. Note that the reconstructed signal is distorted at both ends as a result of the window shape and the overlap-add method, but is an exact copy of the unwindowed signal where the overlap is complete (i.e., between 8 and 32 time units).

Conclusions Figure 6 shows the result of a *real time* overlap-add deconvolution experiment with the Variform. After a startup interval that is related to the window size and amount of window overlap, the tracking error is reduced to nearly zero. Machining experiments will begin in the near future to obtain practical validation of both deconvolution techniques. A small, concave, off-axis sphere machined into an otherwise flat surface has been selected for these tests. This surface was chosen because the command trajectory is aperiodic and its frequency content can be changed by modifying the off-axis distance and/or spindle velocity. Most important though is that the *f-number* of the spherical surface can be selected so that the results are easily characterized with an interferometer.

Reference

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¹ A window function is a vector of weighting coefficients which are zero everywhere outside of a certain interval.

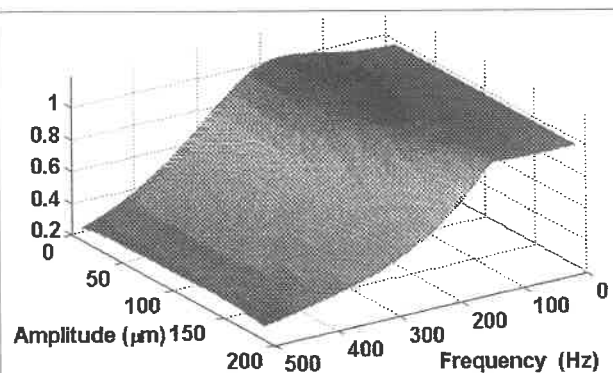


Figure 4. Expanded frequency response

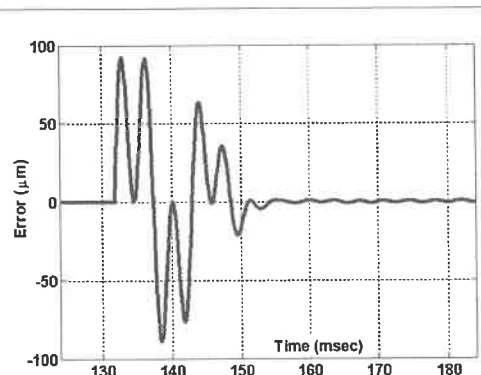


Figure 6. Real time deconvolution tracking error for a 90 μm amplitude command with 90 Hz and 270 Hz sinusoidal components

Machining of Freeform Optical Surfaces by Slow Slide Servo Method

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Introduction

In recent years a significant amount of work has been accomplished in the area of freeform optical surface generation. Most of this work is driven by market demand for these types of surfaces, which currently includes eyewear, electro-optics, defense, automotive, and others. Presently there are several methods to manufacture such surfaces of which the most common ones are grinding and fly cutting. Both grinding and fly cutting rotate the tool and traverse either the tool or part in three linear axes to cut the surface. Grinding and fly cutting can produce very accurate surfaces but require long machining cycles and are difficult to set-up. Another method of fabrication is the fast tool servo (FTS), which is widely used in the contact lens industry. Some of the work on FTS dates back to 1980's, Meinel [1] was able to successfully produce phase corrector plates, and Luttrell [2] produced off-axis conic surfaces and tilted flats with the FTS. However, the FTS system has travel limitations; most systems have a maximum travel range under 1mm.

This paper presents an alternative method of freeform optical surface fabrication, the Slow Slide Servo. The Slow Slide Servo is similar to the FTS in that, the part is mounted on the spindle and as it rotates the tool oscillates in and out to produce a surface. Unlike the FTS method this system does not use any additional axes to oscillate the tool; the Z-axis slide generates the oscillations. Another difference is the spindle position control or C-axis. In an FTS set-up the spindle has an encoder that feeds the position to the FTS unit without putting the spindle in position control. In a Slow Slide Servo all axes are under fully coordinated position control. The Slow Slide Servo can oscillate at ranges in excess of 25mm, it is easy to set-up, it is inexpensive, and it can produce very accurate parts.

System Specification

To implement the Slow Slide Servo method several key features must be available on the diamond turning lathe. Most of these features are the same for both the linear and the rotary axes. They include friction free bearings that generate little heat, direct drive motors with no mechanical compliance between the motor and the feedback system, high-resolution encoders, and minimal structural dynamics in the control loop. Another key feature is the control system or CNC. The CNC must have high-speed data processing, look-ahead capability, a high-resolution data acquisition system and high-order trajectory generation.

The Moore Nanotechnology 350UPL was used for the work presented in this paper. This diamond turning lathe has a T-shape configuration with the spindle mounted on the X-axis and the tool on the Z-axis. The spindle can operate in two separate modes, velocity mode and position mode. The spindle is used in velocity mode for typical axisymmetric diamond turning work with a maximum speed of 6,000 RPM. In the position mode the spindle uses an optical encoder to close the position loop. The same actuator motor and amplifier is used for both configurations. The resolution of the encoder and its electronics is 0.063 arc-seconds or 20,480,000 counts/rev. The C-axis positioning accuracy is ± 2 arc-seconds. In this mode the spindle can operate at a maximum speed of 2,000 RPM. In order for the machine to operate as a Slow Slide Servo careful analysis of the position loops is required.

Axes Performance

A key requirement for the success of the Slow Slide Servo is the implementation of high closed position loop bandwidth on all axes. The bandwidth of a system is an indication of how well the system will respond in the time domain [3]. A system with low position loop bandwidth will have a sluggish response and will also exhibit a large time delay between the commanded position and the actual position. This time delay is an indication of phase shift. An ideal system would have zero phase shift.

A block diagram of the C-axis control loop is shown in Figure 1. The diagram shows a PID (Proportional-Integral-Derivative) loop with velocity and acceleration feedforward. The block containing $B(s)/A(s)$ is the transfer function between the amplifier command and the encoder output.

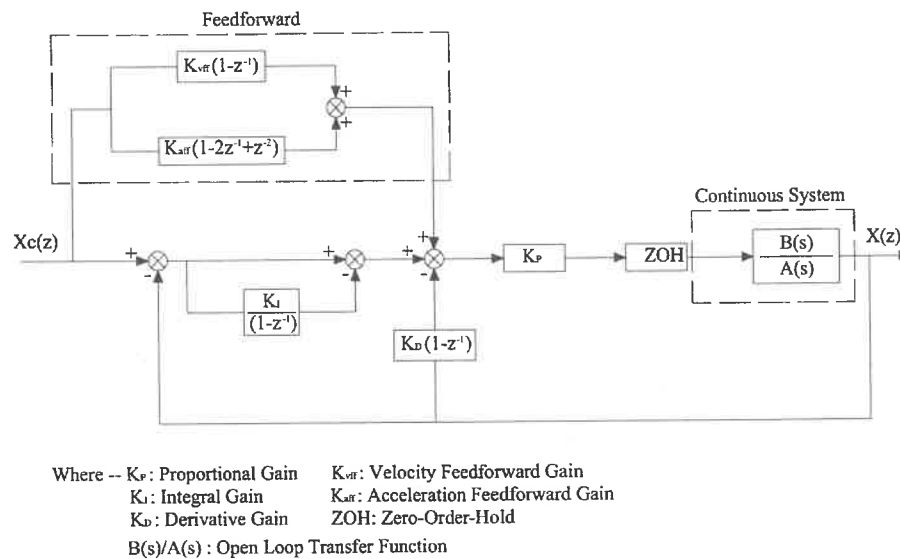


Figure 1. C-axis overall block diagram.

This transfer function includes motor and amplifier electrical dynamics and all associated structural dynamics in the spindle (shaft, encoder, and motor...). The PID loop is tuned for the highest gain values. Figure 2 shows the closed position loop transfer function of the system. The transfer function indicates that the system has a 170 Hz bandwidth (Measured at -3dB crossing). Further, it shows that no significant modes exist in the spindle structure below 1000 Hz. The first destabilizing mode is at 1100 Hz, which is well attenuated with a second order low pass filter. Overall the transfer function shows that the system behaves like a second order system.

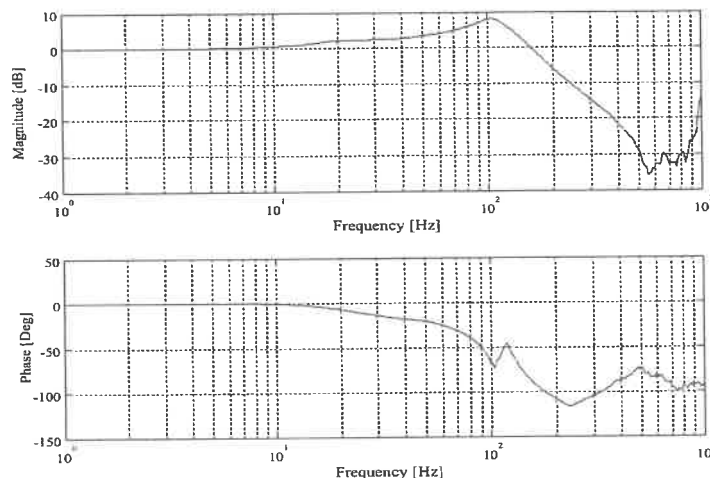


Figure 2. Closed position loop transfer function.

The phase plot shows that the system has zero phase up to a frequency of 10 Hz, after that the phase starts decreasing. Zeroing the phase across the entire bandwidth is a very challenging task, because

a causal system cannot have zero phase. To correct for the inherent phase lag, feedforward terms must be added (Figure 1). The feedforward terms will minimize the tracking error or the time delay between the commanded and actual position by altering the trajectory generation. The velocity and acceleration terms modifies the velocity and acceleration profile of the commanded trajectory. This reduces the tracking error to near zero.

Since the system is a direct drive without any mechanical advantage, all the stiffness has to come from the electrical dynamics. Measuring the static compliance of the system shows that it is infinitely stiff. This is due to the integral term in the PID loop. The integral gain corrects for any static error. The dynamic compliance in Figure 3 shows the system compliance at any particular frequency. The system stiffness is below 1 arc-sec/in-lbs. for up to 10 Hz. The worst stiffness occurs at the natural frequency and is less than 2.5 arc-sec/in-lbs.

Furthermore, the C-axis was commanded five step moves in both the negative and positive direction to show its response to small step commands. Figure 4 shows the response to quarter arc-second commands.

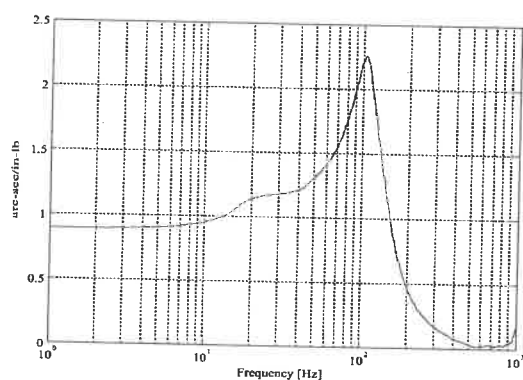


Figure 3. C-axis compliance.

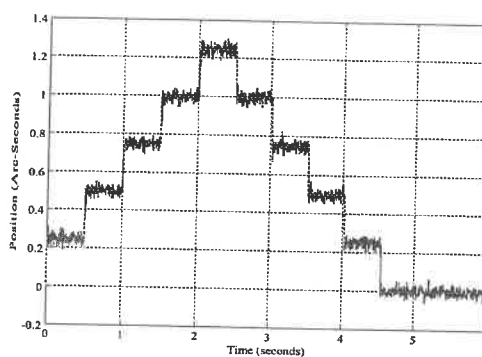


Figure 4. C-axis step response.

The X-axis and Z-axis of the machine have comparable performance to the C-axis. They both have a 100 Hz position loop bandwidth. In addition optimized acceleration and velocity feedforward gains are added to minimize tracking errors.

Machining Results

Several different freeform surfaces have been machined using the Slow Slide Servo to show the viability of this type of machining. One example of these surfaces is the cubic phase plate shown in Figure 5. This part was machined using Zinc Sulfide with a negative rake diamond tool. The sag of the surface is 100 μ m PV. The form results shown in Figure 6 indicate that the PV error is 0.26 μ m and the surface finish shown in Figure 7 averages a roughness of 4.6nm.

A second example is the machining of an off axis sphere. A 75mm diameter 6061 aluminum concave sphere with a 75mm radius is offset from the spindle center by 7.686mm. The sphere is then cut with a 1.5mm radius diamond tool. The maximum oscillation of the Z-axis is approximately 11mm. The form results of the sphere are shown in Figure 8. It shows that the sphere form accuracy is 0.33 μ m PV. The finish result shown in Figure 9 has a roughness average of 5.6nm. Several other surfaces have been machined including tilted flats, progressive lenses, cylinders, torics, and biconics in a variety of materials. They all showed comparable form and finish results to the above samples.

Conclusions

Most of the machining tests performed with the Slow Slide Servo indicates that it is a very viable method for producing freeform surfaces. Surface finish and form accuracy results are comparable to

axisymmetric diamond turning results. In addition, this method is inexpensive, does not have sag limitations, is very accurate, and is easy to set-up.

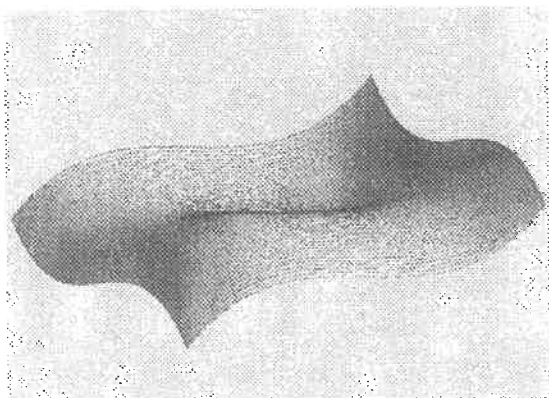


Figure 5. Surface of cubic phase plate.

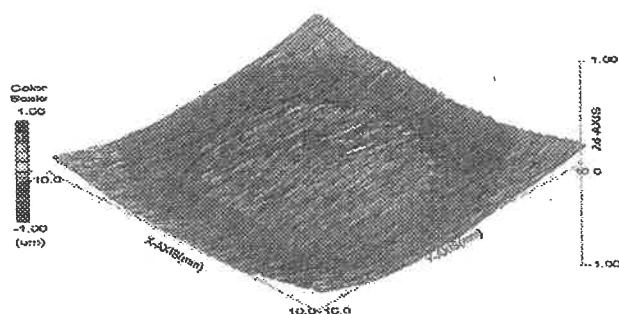


Figure 6. Phase plate form accuracy results.

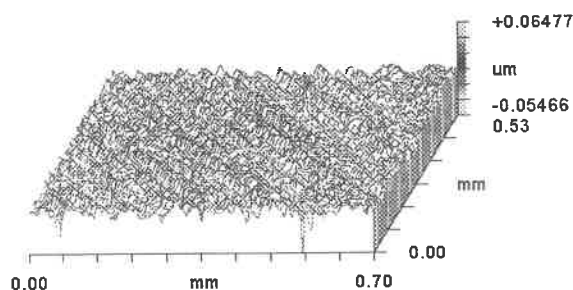


Figure 7. Phase plate surface finish results.

Cubic Phase Plate
Material: Zinc Sulfide 27mm X 27mm

Form results (Panasonic UA3P)
PV-0.263 μ m
Rms-0.055 μ m

Finish results (Zygo NuView)
Ra-4.569nm
Rms-6.077nm

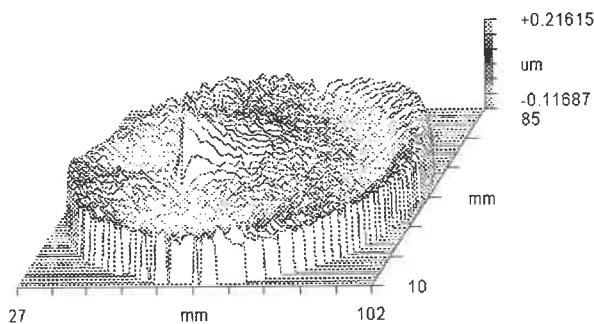


Figure 8. Off-axis sphere form results,
PV-0.333 μ m, rms-0.047 μ m.

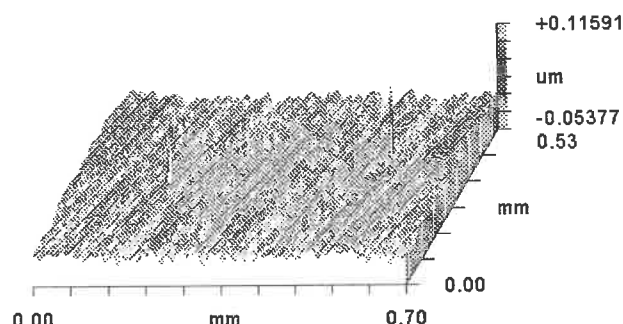


Figure 9. Off-axis sphere finish results,
Ra-5.455nm, rms-6.753nm.

Acknowledgments

The authors wish to thank the *Center of Optics Manufacturing* for their assistance with the cubic phase plate measurement.

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High Bandwidth Short Stroke Rotary Fast Tool Servo

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Keywords: fast tool servo, rotary fast tool servo, high bandwidth, non-axisymmetric, textured surface, diamond turning, National Ignition Facility.

This paper presents the design and performance of a new rotary fast tool servo (FTS) capable of developing the 40 g's tool tip acceleration required to follow a 5 micron PV sinusoidal surface at 2 kHz with a planned accuracy of 50 nm, and having a full stroke of 50 micron PV at lower frequencies. Tests with de-rated power supplies have demonstrated a closed-loop unity-gain bandwidth of 2 kHz with 20 g's tool acceleration, and we expect to achieve 40 g's with supplies providing ± 16 Amp to the Lorentz force actuator. The use of a fast tool servo with a diamond turning machine for producing non-axisymmetric or textured surfaces on a workpiece is well known. Our new rotary FTS was designed to specifically accommodate fabricating prescription textured surfaces on 5 mm diameter spherical target components for High Energy Density Physics experiments on the National Ignition Facility Laser (NIF).

The basic topologies for a rotary fast tool servo and a linear fast tool servo are shown in **Figure 1**. A rotary fast tool servo is preferred in certain diamond-turning applications that are intolerant to the reaction force developed by a linear fast tool servo. For instance, in the present case where it is desired to produce a textured surface on a spherical workpiece a fast tool servo is mounted on a relatively slow rotary axis (B-axis) that allows the tool to engage the workpiece at all points from its "pole" to its "equator", as depicted in **Figure 2**. A rotary-type mechanism oriented with its rotation axis parallel to the B-axis generates a reaction torque on the B-axis, which can be allowed to float as a reaction mass or be locked and allowed to transmit the torque to the machine structure. In the latter case, the machine structure experiences a disturbance torque whose value does not depend on the angle of the B-axis. In contrast, a linear fast tool servo will generate a reaction force on the B-axis. This is generally not a problem when the B-axis is positioned so that the reaction force is parallel to the direction of travel of the slide carrying the B-axis. In that case the relatively heavier slide carrying the B-axis acts as a reaction mass to the linear fast tool servo. However, when the B-axis is positioned so that a component of the reaction force is perpendicular to the direction of travel of that slide, that force component is transmitted by the slide to the machine structure as a disturbance. To the extent that the tool/workpiece interaction is affected by disturbances to the machine structure, unless the reaction force is dealt with, a linear fast tool servo will thus produce variations in the desired surface texture as a function of "latitude" on a spherical workpiece.

The above discussion is quantified by comparing a conservative total moving mass rotary inertia of 100 gm-cm² for a rotary FTS with a 5 mm swing radius (twice the inertia of our present system) and an optimistic total moving mass of 10 gm for a linear FTS. A future design goal is to produce a fast tool servo with the 1000 g's tool acceleration needed to follow a 5 micron PV sinusoidal surface at 10 kHz. In this case the reaction torque from the rotary FTS and reaction force from the linear FTS are 20 N-m and 100 N, respectively. If the FTS is mounted on a B-axis that is allowed to float, a reaction torque causes an angular acceleration of the rotating element of the B-axis creating a tangential displacement between the surface of a spherical workpiece and the tool. Using a conservatively low rotary inertia of 0.0052 kg-m² for the moving element of the B-axis (e.g. a non-motorized 4 inch air bearing spindle), a 20 N-m rotary FTS reaction torque produces a negligible ± 5 nm tangential motion between the tool and a 5 mm diameter spherical workpiece. In comparison, without an effective reaction mass between the linear FTS and the B-axis, the 100 N reaction force is transmitted directly through the B-axis as a disturbance to the machine structure, which is surely a large disturbance.

Our new high bandwidth rotary fast tool servo is shown in **Figure 3**. The swingarm carries a diamond tool at a swing radius of 5 mm and has a full-travel angle of ± 5 mrad. The small rotation angle allows using over-constrained crossed-flexures to guide the rotary motion. A total of eight flexures are used, four above the tool and four below it. The inner ends of the flexures are fixed to the swingarm, and the outer ends are fixed to the base. To ease manufacturing and assembly, pairs of collinear flexures are formed from a single piece, as best seen in **Figure 4**. Each flexure is a 0.15 mm thick piece of 1095 blue tempered spring steel that has been polished along its long axis to minimize initiation of fatigue cracks. Each flexure is pre-tensioned with a 50 N load before clamping the ends. This pre-tension, along with the over-constrained arrangement, is intended to provide a well-defined axis of rotation that does not wander when the swingarm is rotated. The measured stiffness at the plane of the tool along the

centerline is 22 and 15 N/micron in the axial and radial directions, respectively. The 5 mm swing radius combined with a moving mass rotational inertia of 51 gm-cm² results in an effective linear mass of 0.2 kg as seen at the tool tip. The expected dynamic stiffness of the tool at 2 kHz is on the order of 20 N/micron. The parasitic transverse motion of the tool tip is only +/- 62 nm for the full travel of +/- 25 micron, making it negligible and thus simplifying CNC programming for machining a workpiece. A drawback of using a small swing radius is that the actuator acceleration is de-magnified as reflected into tool tip acceleration.

Our new rotary fast tool servo uses a commercially available small diamond tool weighing 0.4 gm (diamond plus steel shank), shown in Figure 4. The sides of the tool shank form a 30 degree taper angle that is received by a tapered slot in the swingarm. To insure a high stiffness connection between the tool and swingarm, the sides of the tool shank are relieved to provide four distinct areas of contact with the swingarm, two in the front and two in the rear. A leaf spring is used to transmit the hold-down force from a setscrew to the center of the tool shank.

We used a commercially available moving-magnet galvanometer for the actuator, and built our own power amplifier and current-loop compensation circuit. Figure 5 shows a block diagram of the control system and the measured small signal closed-loop response of the tool position. The inner current loop has a crossover frequency of 30 kHz, essentially removing the electrical dynamics of the power amplifier and actuator at frequencies up to the desired 2 kHz. We designed and used an integral rotary viscous damper consisting of a thin circular plate attached to the bottom of the swingarm. A 0.1 mm gap between each face of the circular plate and the adjacent stationary surface is filled with heavy grease. This damper significantly reduced the effect of a 5 kHz resonance between the swingarm and the moving element of the actuator, making it possible to use a simple PI controller for stabilizing the outer tool position loop with a crossover frequency of 1 kHz. The inner loop was implemented with analog components, while the outer loop compensation is implemented with a PC-based digital controller using an 80 kHz sampling rate. The position of the tool is measured by a pair of eddy current sensors operating differentially and looking at the back of the swingarm on either side of the axis of rotation and near the elevation of the tool. This is best seen in Figure 6, which shows the swingarm and diamond tool engaging a workpiece during the first cutting test. The eddy current sensor amplifier produces a +/- 10 V signal for the +/- 25 micron tool swing, and has a noise level of 4 mV PV which corresponds to +/- 10 nm PV.

During the first cutting test shown in Figure 6 the tool was commanded to hold a constant position while facing the workpiece, so that the robustness of the FTS mechanism and the control system could be benchmarked. We used a diamond tool with a 25 micron nose radius and a federate of 2.5 micron/rev. This leads to a rather coarse theoretical finish of 30 nm PV, but since we were exercising fairly loose control over the thermal environment at the work zone we chose to use a relatively fast feed rate to better assure a constant depth of cut across the 9 mm diameter part. The resulting surface on the 6061-T6 aluminum workpiece (not the most ideal material to diamond turn, but readily available) is also shown in Figure 6. Preliminary measurements indicate that the diamond turning groove depths match the theoretical 30 nm PV, but are superimposed on a 60 nm PV waviness with a 30 micron spatial wavelength, resulting in a 13 nm rms roughness for a 0.18 mm scan length. The 30 micron waviness corresponds to a 0.7 Hz disturbance. Likely culprits are least-bit dithering of the non-locked in-feed slide, mechanical ground noise through the passive air isolators, and spindle air pressure fluctuations.

Additional cutting tests will be performed to determine the ability of our new rotary FTS to produce sinusoidal surfaces on the face and diameter of a workpiece at 2 kHz. Follow-on work includes implementing advanced controller algorithms such as Feedforward cancellation to account for phase delay at higher frequencies and Adaptive Feedforward Cancellation to reduce following error and reject disturbances at select frequencies. We also plan to design, build, and test a new prototype intended to provide the same stroke at 10 kHz.

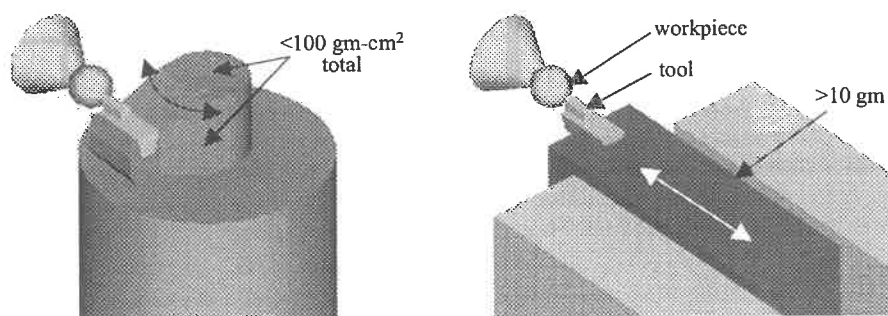


Figure 1: Basic topologies for Rotary (left) and Linear (right) fast tool servos.

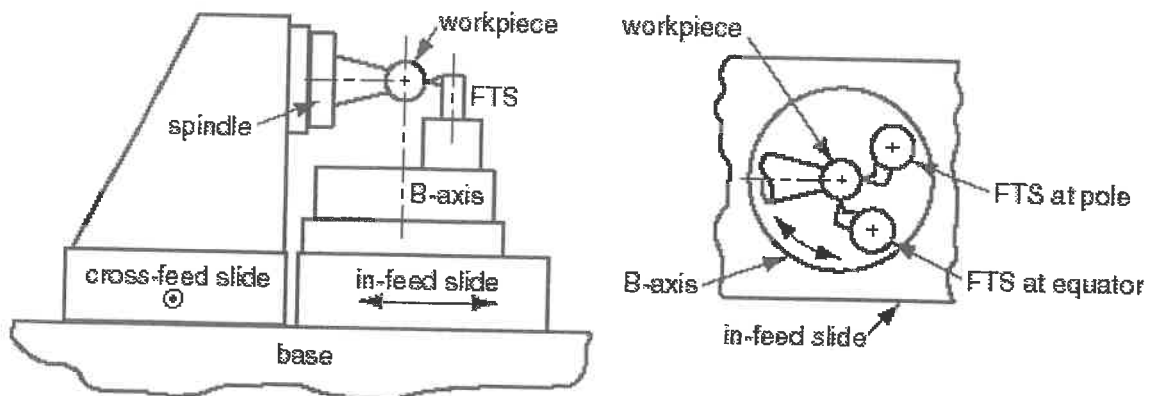


Figure 2: Sketch of a fast tool servo set up on a two-axis lathe with a B-axis for machining a spherical workpiece. Side view (left). Top view (right) showing tool at equator and at pole of workpiece.

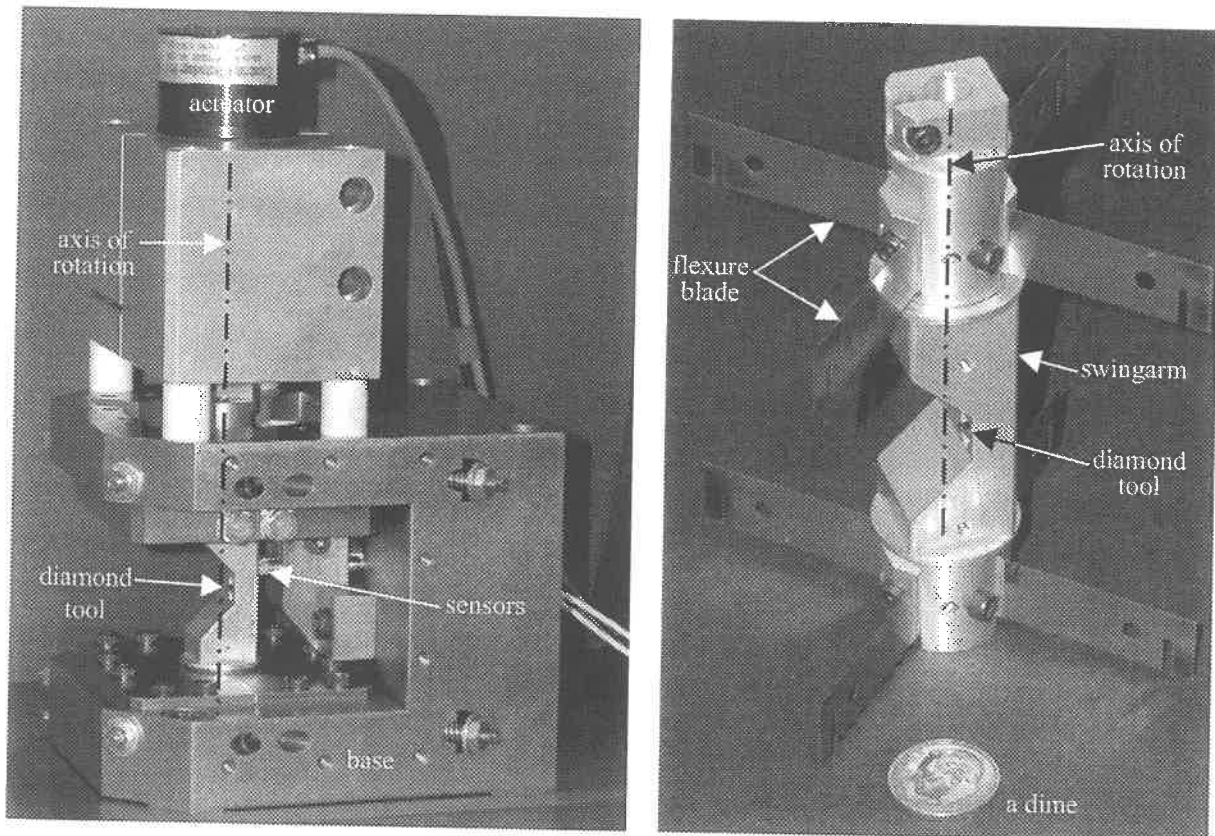


Figure 3: Our 2 kHz rotary fast tool servo (left), and its rotating element not including the actuator (right).

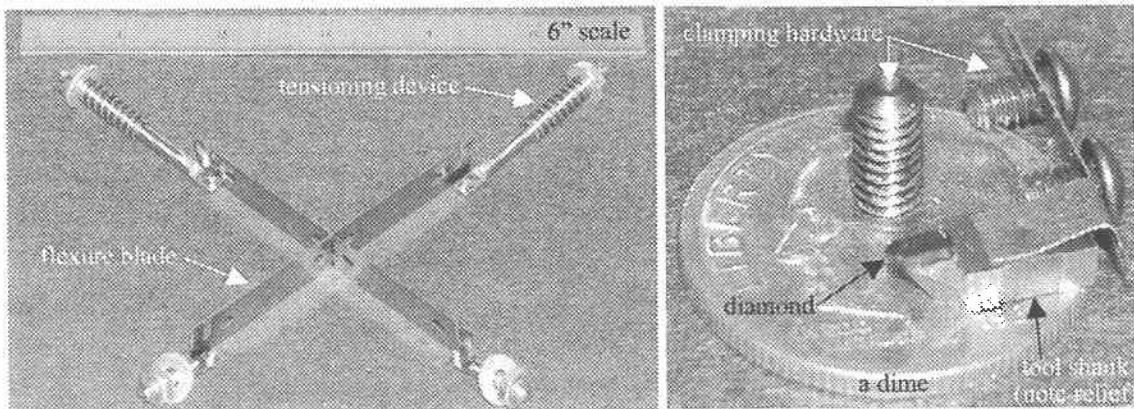


Figure 4: Four flexure blades and tensioning devices (left), and diamond tool with clamping hardware (right).

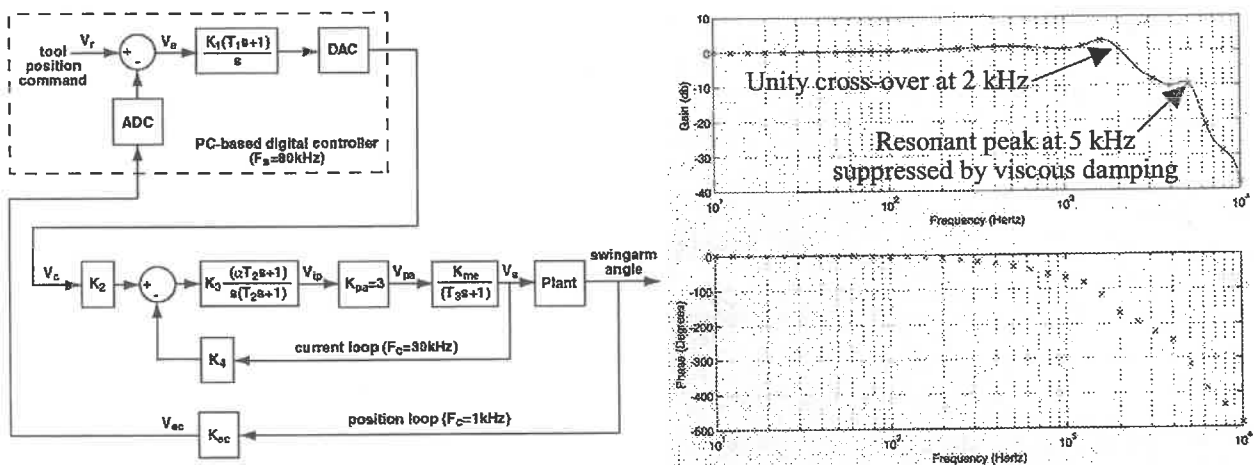


Figure 5: Control system block diagram (left), and measured small-signal closed-loop response (right).

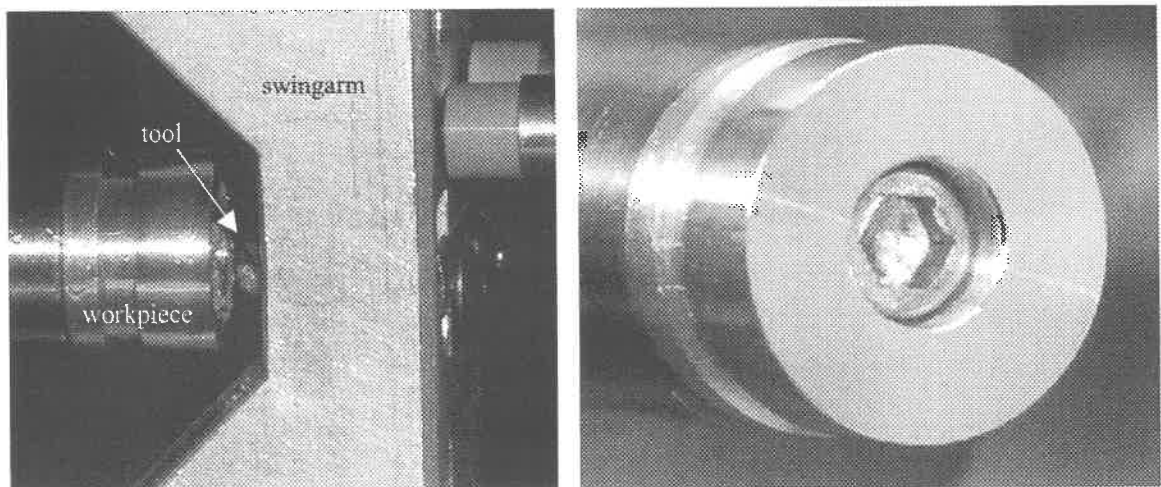


Figure 6: First cutting test with our 2 kHz rotary FTS. Side view (left) and face view of part (right).

FABRICATION OF A MICROROBOT MOVABLE IN FLEXIBLE PIPES LIKE THE LARGE INTESTINE

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1. Introduction

An inspection of the large intestine is very effective in order to prevent the large intestine cancer. A fiberscope is excellent for the inspection of the large intestine. However, the inspection by the fiberscope often attends many pains and injuring of the intestine. We are hoping the inspecting method for the intestine with no pain and no injuring of the intestine before the operation by the fiberscope. Some microrobots which can move in the large intestine have been proposed by several research groups [1], [2]. However, the sure moving in the large intestine has not been confirmed, because the actual large intestine in our human body is especially very flexible to the radius direction and the mobile moving in the flexible pipes is very difficult.

Now, we propose a new model of mechanism that can surely move in a flexible pipe like the large intestine. We need holding the flexible pipe to the radius direction and driving the mechanism to longitudinal direction using the friction force by the holding the flexible pipe in order to move surely in the flexible pipe. We use three rubber bellows in series. Two outer rubber bellows are provided eight bulging rubber sheets. These are called as holding elements with braking mechanisms. So, the holding element works as a break of the moving mechanism. The center rubber bellows is called as a driving element, because the rubber bellows drive the moving mechanism by its stretching and shrinking motion.

When the rubber bellows of the holding element shrinks, eight bulging rubber sheets

spread to the radius direction and touch the flexible pipe. Then the spread bulging rubber sheets surely hold the flexible pipe. When the rubber bellows of the holding element stretches, spreading to the radius direction of the eight bulging rubber sheets and touching to the large intestine are canceled. The rubber bellows are stretched by pneumatic pressure and shrank by vacuum pressure. The stretching and shrinking of the rubber bellows are controlled by a computer and electromagnetic valves.

We confirmed that the new mobile mechanism, which is equipped two holding elements and a moving element, can move to the forward and backward direction in the flexible pipe like the intestine.

2. Structure of the fabricated mobile moving mechanism

Structure of the fabricated mobile moving mechanism is shown in Fig. 1. The mobile moving mechanism consists of two holding elements and a moving element. These elements are driven by three bellows which are 16 mm in outer diameter, 11 mm in inner diameter, 30 mm long and made of nitrile butyl rubber (NBR). Each bellows is an independent vessel. Three air-feeding tubes, which are 1.2 mm in outer diameter, 0.6 mm in inner diameter and 1.5 m long, are connected to three bellows of the holding elements and the moving element. The rubber bellows are stretched by the supply of the pneumatic pressure and are shrunk by the supply of the vacuum pressure. The stretching and shrinking of the bellows are controlled by three electromagnetic valves and a computer.

A braking mechanism of the holding element is shown in Fig. 2. The braking mechanism consists of eight bulging rubber sheets which are 5 mm wide, 0.5 mm thick, 20 mm long and made of NBR. The length of the braking mechanism is 17 mm in the free condition and 14 mm when the pressure in the bellows is vacuum. The diameter of the braking mechanism becomes to about 26 mm. Then the bulging rubber sheets are strongly pressed to the flexible pipe like the large intestine. The friction force in the flexible pipe are made by the radial force of the bulging rubber sheets.

3. Experimental apparatus

An experimental apparatus for measuring the characteristics of the mobile moving mechanism is shown in Fig. 3. A computer controls three electromagnetic valves through a valve controller. Three air-feeding tubes are connected from the electromagnetic valves to two holding elements and the moving element of the mobile moving mechanism. An air-compressor is connected to the entrance ports of the electromagnetic valves and feeds pneumatic pressure to stretch the bellows. A vacuum pump is connected to the exit ports of the electromagnetic valves and feeds vacuum pressure to shrink the bellows.

4. Moving principle of the microrobot

The pneumatic and vacuum pressure are fed like as a time-chart shown in Fig. 4. The mass flow rate through the electromagnetic valve is proportional to the absolute pressure of the upstream flow, because the electromagnetic valve is a kind of an orifice. The absolute pneumatic pressure at the time of stretching motion is more than two times of the pressure in the bellows in the time of shrinking motion. Then the supplying time of the vacuum pressure in the shrinking motion ($=\tau_2$) must be more than two times of the supplying time of the pneumatic pressure in the stretching motion ($=\tau_1$). The sum of τ_1 and τ_2 is called as the cycle time ($=T$).

As shown in Fig. 4, we make a time difference of $\tau_1/2$ in the pneumatic supplying time among the front holding element, the

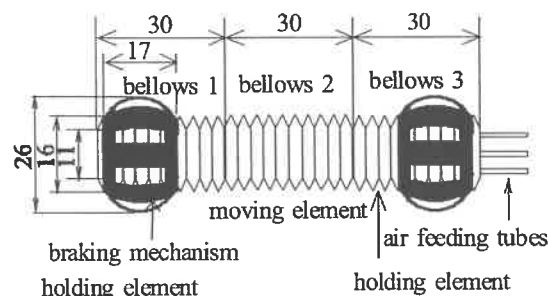


Fig. 1 Structure of the fabricated microrobot

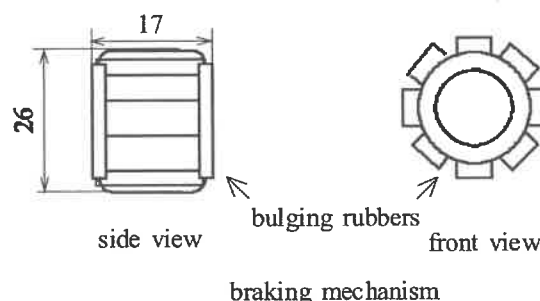


Fig. 2 Structure of a braking mechanism of the hold element

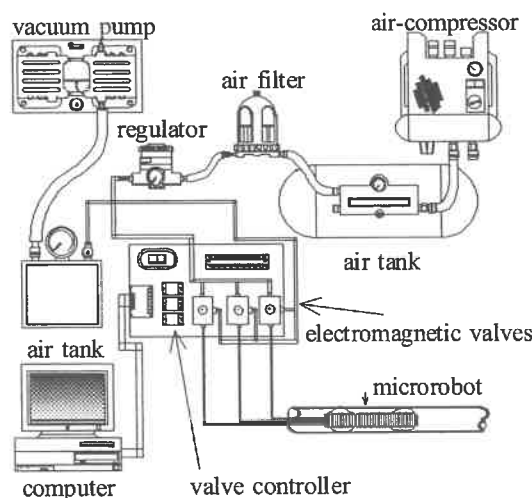


Fig. 3 Experimental apparatus

central moving element and the rear holding element. The mobile moving motion is estimated as Fig. 5, when the pneumatic and vacuum pressure are supplied as shown in Fig. 4.

(1) At initial, all the bellows are shrinking by the vacuum pressure. The braking mechanisms of the front and rear holding element are at the condition of the braking and hold the flexible pipe.

(2) The pneumatic pressure is fed to the front holding element and the braking is free.

(3) The pneumatic pressure is fed to the central moving element and the mobile moving mechanism is stretching. Then the front part of the mobile moving mechanism can move to the forward direction, because the rear holding element is still at the condition of the braking and holds the flexible pipe.

(4) The pneumatic pressure is fed to the rear holding element and the braking is free. At the same time, the vacuum pressure is fed to the front holding element and it is at the condition of the braking. The vacuum pressure is fed to the central moving element and the mobile moving mechanism is shrinking. Then the rear part of the mobile moving mechanism can move to the forward direction, because the braking of the rear holding element is free.

(5) All the bellows are shrunk by the vacuum pressure. The braking mechanisms of the front and rear holding element are in the condition of braking and hold the intestine. It is same as the initial condition and one cycle is over. The mobile moving mechanism moves the stretching displacement of the moving element. Consequently the speed is shown by (stretching displacement) / (cycle time).

5. Measurement of the speed

We measured the speed of the mobile moving mechanism supplying the pneumatic pressure of 0.08 [MPa] and the vacuum pressure of -0.08 [MPa]. We chose that the supplying time of the pneumatic pressure in the stretching motion ($=\tau_1$) is 0.2 seconds. The supplying time of the vacuum pressure in the shrinking motion ($=\tau_2$) was changed.

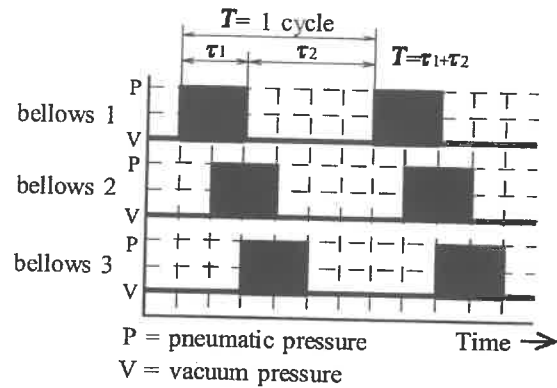


Fig. 4 Time chart of the pneumatic and vacuum pressures for the each bellows

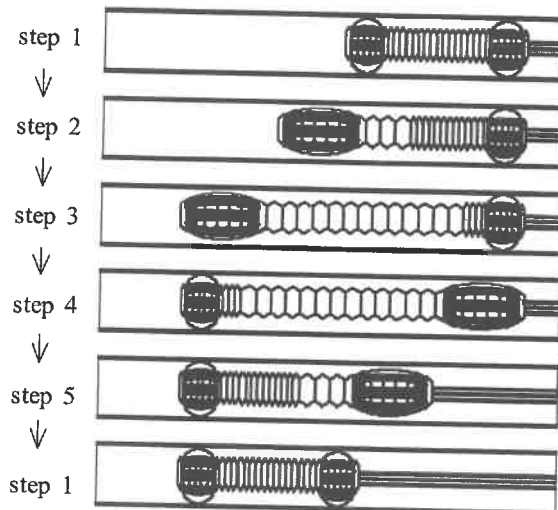


Fig. 5 Moving principle of microrobot

Relationship between the cycle time and the speed is shown in Fig. 6. The mobile moving mechanism was confirmed to move at the speed of 39 [mm/s]. In this experiment, it was also confirmed that the mobile moving mechanism is able to move to the vertical direction and to generate the traction force of 0.8 [N].

6. Conclusions

(1) We fabricated of a new microrobot that can surely move in a flexible pipe like the large intestine. The moving mechanism consists of two holding elements and a moving element. The holding element has a braking mechanism which consists of eight bulging rubber sheets.

(2) We confirmed by the experiment that the moving mechanism can move to the forward and backward direction in the flexible pipe like the intestine. The mobile moving mechanism was confirmed to move at the speed of 39 [mm/s]. It was also confirmed that the mobile moving mechanism is able to move to the vertical direction and to generate the traction force of 0.8 [N].

(3) The mobile moving mechanism may be used to the inspection mobile microrobot for the human large intestine, because the pig's small intestine is very similar to the human large intestine.

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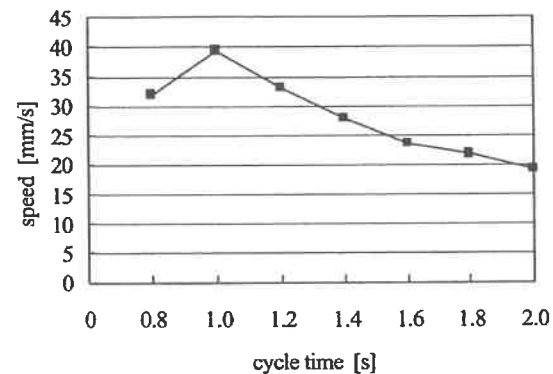


Fig. 6 Relationship between speed and cycle time

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Design and Analysis of Nano-positioning Stage Driven by Piezoelectric Elements

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1. Introduction

A demand for an ultra-precision stage in the field of nanotechnology has recently been increased, especially in AFM applications. Larger sized wafers require a stage to have fast response as well as long travel range maintaining high accuracy. Piezoelectric elements-driven ultra-precision stages have been used for high accuracy, fast response and high load capacity, which are allowable to apply the stages to the AFM applications.

Most of the piezoelectric elements-driven stages are guided by flexure hinges for force transmission and mechanical amplification^(1,2). However, the flexure hinge mechanisms cause lack of position accuracy due to coupled and parasitic motions. Therefore it is important that the mechanism design of the stage is focused on the stiffness of the flexure hinges to accomplish fast response and high accuracy without coupled and parasitic motions.

In this study, we present an optimal design of a piezoelectric elements-driven precision stage with two DOF. For the optimal design, a cost function is minimized by a Min-Max design algorithm⁽³⁾, subject to design constraints. Also, FEM analyses are carried out to certify the design requirements. Finally, experimental results show the validity of the designed mechanism.

2. Nano-positioning stage

The configuration of a nano-positioning stage driven by piezoelectric elements is shown in Fig. 1. The stage has the structure of a monolithic nested-loop type moving plates, in which each moving plate is guided by two four-link mechanisms. The four-link mechanism plays roles of motion guidance for translation as well as transmission of a preload to the piezoelectric element. The four hinges of the four-link mechanism are implemented by four round-notched flexure hinges. Due to the nested-loop type structure, the moving plates are actuated independently each other by piezoelectric elements so that the stage can avoid coupled interference motions.

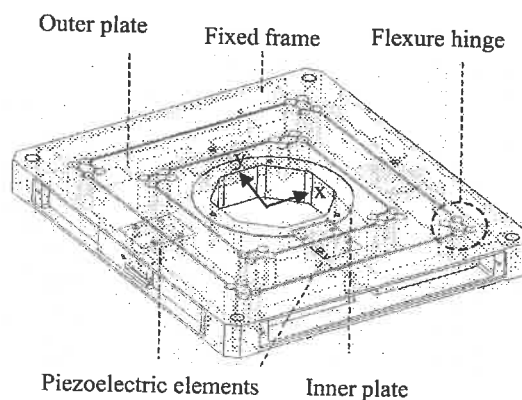


Fig. 1 Structure of two-axis nano-positioning stage

3. Mathematical model

The flexure hinges in the four-link mechanism deform along both axial and rotational directions

when the plate moves along translational direction. However the axial deformation of the flexure hinges are so small compared to the small translation of the plate that they may be ignored. Assuming that the flexure hinges operate as rotational springs, the inner and the outer plates are rigid bodies, and dimensions of the four four-link mechanisms are the same. The masses of the inner and the outer plates are M_1 and M_2 . In the four-link mechanism, the distance between two rotational joints connecting the rotational link is L , and the mass and the inertia moment of the rotational link are m and I . The equations of motions are decoupled as follows:

$$\left(M_1 + M_2 + 5m + 4\frac{I}{L^2}\right)\ddot{x} + 8\frac{k_\theta}{L^2}x = 0 \quad (1)$$

$$\left(M_2 + m + 4\frac{I}{L^2}\right)\ddot{y} + 8\frac{k_\theta}{L^2}y = 0 \quad (2)$$

where k_θ is the rotational stiffness of the flexure hinge.

The rotational stiffness of the round-type flexure hinge, k_θ , is given by

$$k_\theta = \frac{2Ebt^{5/2}}{9\pi r^{1/2}} \quad (3)$$

where E , b , r and t are Young's modules, the thickness of the plate, the radius of the hole and the distance between two holes.

4. Design of nano-positioning stage

This stage is to measure the critical depth (CD) of nano patterns on a silicon wafer. For this task, the stage requires specifications as listed in Table 1. In Table 1, the working range and the first resonance frequency must be considered in design indices.

The angular errors consist of parasitic motion error, a machining error and an assembly error. The parasitic motion error can be investigated through

simulation of the mechanism, however the machining error and the assembly error can be measured after machining and assembling the stage.

Table 1 Required specifications

Working range	25 μm $\times$ 25 μm
Resolution	5 nm
First resonance frequency	≥ 200 Hz
Angular error	≤ 50 μrad

The resolution of the stage is decided by the resolution of position sensor, the input and output resolution of the amplifier for a piezoelectric element, a control algorithm and the resolution of D/A and A/D converters. Therefore the specifications both of the first resonance frequency and the working range cause following constraint conditions:

$$f_1 \geq 200 \text{ Hz} \quad (4)$$

$$K \leq \frac{1}{10} k_{pzl} \quad (5)$$

$$\sigma_{\max} \leq \frac{1}{10} \sigma_Y \quad (6)$$

where f_1 , K , k_{pzl} , $\sigma_{\max}$ and σ_Y are the first resonance frequency, the translational stiffness of the stage, the stiffness of the piezoelectric element, the maximum stress and the yield stress, respectively. To simplify the design problem, let the parameters M_1 , M_2 , m , I and b be fixed. Then, the hole radius r and the distance t between the holes are obtained by a design algorithm in following additional constraint conditions:

$$r_1 \leq r \leq r_2 \quad (7)$$

$$t_1 \leq t \leq t_2 \quad (8)$$

In this paper, the optimal design parameters r and t were obtained by Min-Max design algorithm. Design indices w_1 to w_4 are chosen as follows:

$$w_1 = |f_1 - f_d| \quad (9)$$

$$w_2 = |K - K_d| \quad (10)$$

$$w_3 = r \quad (11)$$

$$w_4 = t \quad (12)$$

where f_d and K_d are desired resonance frequency and desired stiffness. The design indices w_1 , w_2 and w_3 are in favor of the minimum value whereas the design index w_4 is in favor of the maximum value in the constraint conditions. After the design indices are normalized, they are weighted. Then a composite design index W_c can be found as follows:

$$W_c = \tilde{w}_1 \wedge \tilde{w}_2 \wedge \tilde{w}_3 \wedge \tilde{w}_4 \quad (13)$$

where $\tilde{w}_i$ is i -th weighted normalized design index and the symbol $\wedge$ is a fuzzy intersection. The optimal design parameters r and t are the maximal position of W_c in the constraint conditions. In this paper, we obtained the design parameters as

$$(r, t)_{opt} = (4.9, 1.8) \text{ mm} \quad (14)$$

5. Analyses of the designed stage

The resonance frequencies, the stiffness, the maximum stress calculated from the mathematical model using (14) are listed in Table 2. The calculated values satisfy the constraint conditions (4)~(6).

Table 2 Calculated results using mathematical model

f_1, f_2	201, 356 Hz
K	6.0 N/ μ m
σ_{max}	49.8 MPa

Also, the values in Table 2 were compared to the results of FEM analyses. Fig. 2 shows the results of dynamic analyses. Fig. 2-(a) and (b) show that the first and the second modes are in accordance with the

calculated values. Fig. 2-(c) and (d) show that the frames of plates deform in high frequency, which violates an assumption that the plate is rigid.

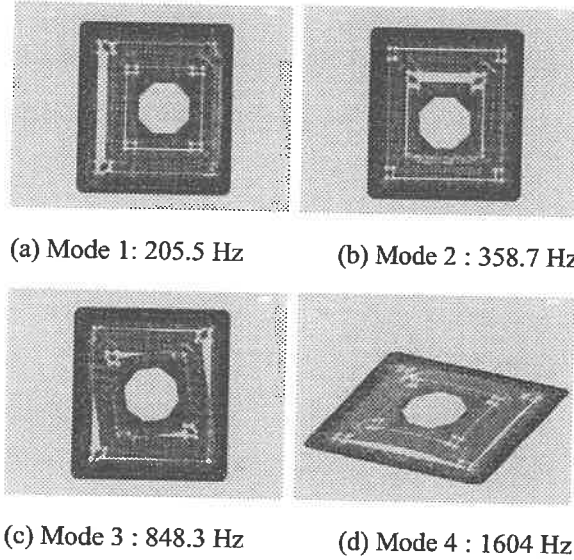


Fig. 2 Dynamic modes

The static analyses of the stage were also carried out. Fig. 3 shows the static deformation shapes applied by forces at x- and y-axis, and the weight of the stage. The stiffnesses based on the deformation and the applied forces are 5.97 N/ μ m and 5.96 N/ μ m, in x- and y-axis, respectively. The stiffnesses of x- and y-axis accord the calculated results in Table 2.

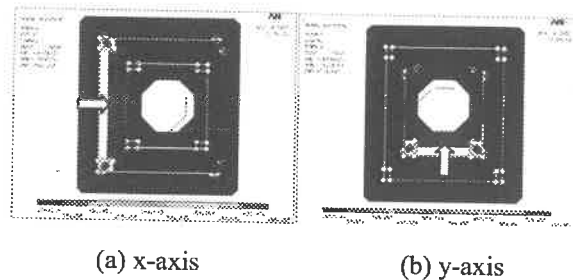


Fig.3 Static deformation

The maximum stress is generated at the flexure hinge when the moving plates arrive at the maximum displacement. The analyzed maximum stress is 42.2

MPa which is within 15 % of the calculated value in Table 2.

6. Experiments

The developed stage is shown in Fig. 4. The first resonance frequency measured without the piezoelectric elements is 197 Hz, which is a little lower than the analyzed one. When the piezoelectric elements are attached at the stage, the first resonance frequency is increased to 212 Hz.

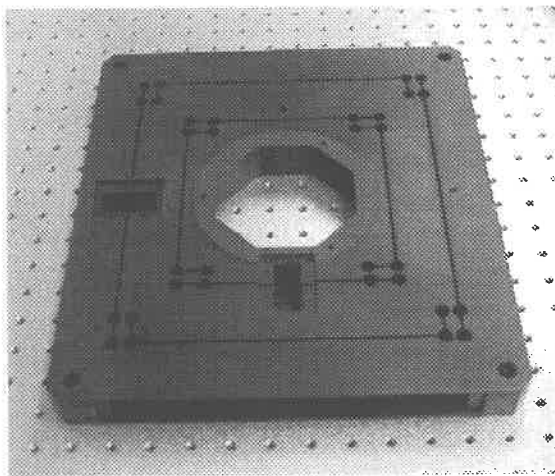


Fig. 4 Developed stage

Fig. 5 shows the stepwise response of x-axis. The steady-state response shows that the resolution of this stage is 5 nm.

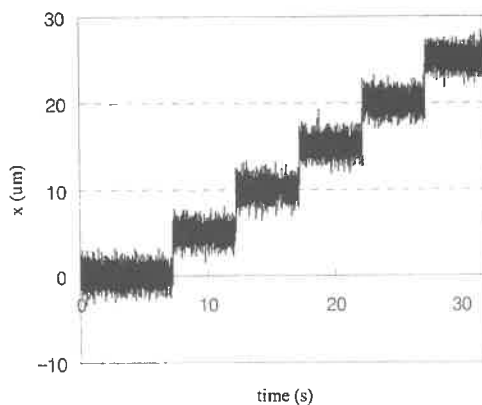


Fig. 5 Stepwise response

The angular errors were measured as shown in

Fig. 6. The angular errors are lower than the required specification.

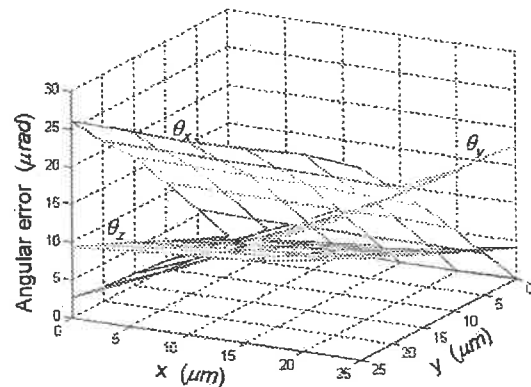


Fig. 6 Measured angular errors

7. Conclusion remarks

The nano-positioning stage for CD measurement was developed. The stage was designed by Min-Max optimal design algorithm, and analyzed by calculations and FEM analyses. The analyzed results satisfied the design requirements. Also, experiments showed the validity of the designed stage.

Acknowledgement

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Theoretical and Experimental Study on Surface Finish for Multi-Axis CNC Milling

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ABSTRACT

This paper theoretically and experimentally studies the effect of machining parameters on surface finish in 3- and 5-axis CNC milling processes. Based on Z-map method, an analytical model has been proposed to simulate surface finish generated by the milling process. Experiments have been carried out to verify the simulation results. The difference between the simulation and experimental results has been discussed and the reason behind the difference is explored.

Keywords: 3-/5-Axis CNC Machining; Simulation; Average Surface Roughness.

1 INTRODUCTION

Advancement in manufacturing technology has led to the development of various techniques to improve the manufacturing process. Recently, computer aided manufacturing (CAM) is used to improve the productivity and quality of manufactured product. Complicated products such as dies and molds can easily be manufactured with the help of advanced CAM systems. These products require high quality surface finish. Milling such as end or ball-end milling is extensively used to manufacture and finish these products. Hence, it is of great importance to predict and control products' surface finish during milling process. Most of the currently available commercial CAM software cannot predict the surface finish. The surface generated during milling is affected by different factors such as vibration, spindle run-out, temperature, tool geometry, feed, cross-feed, tool path and other parameters. During finish milling, the depth of cut is small. In this research, the effect of vibration and temperature on surface finish is not considered.

Due to the complexity of end or ball-end milling process, accurate geometric or physical simulation presents certain difficulty. Different researches have been conducted in this regard. To name a few, Kim *et. al.* [1] have used analytical model to calculate the scallop height, which can be used for simple geometry and the situation where cutting tool is normal to the workpiece surface. The mathematical model becomes very complicated for multiple tool pass operation and 5-axis machining. Kang *et. al.* [2] used experiments to determine the effects of inclination angle, up and down milling on the surface finish during ball-end milling. From the experiments, they observed that better surface finish could be

obtained for certain inclination angles. Similarly, for 5-axis machining, Kruth and Klewais observed that better surface finish was achieved by tilting the tool in feed direction [3]. Baptista and Simoes [4] experimentally determined that scallop heights in 5-axis CNC with end mill inclined in the feed direction were much smaller than those in 3-axis CNC ball end milling. Although substantial work has been done, in order to predict surface finish, a systematic simulation model is needed. In this study, a simulation system has been developed for 3-and 5-axis CNC milling.

2 ANALYTICAL MODEL

2.1 Assumptions

For an otherwise complex milling process, certain assumptions are made to analytically model the process. Cutting tool is modeled based on its geometry and shape. The cutting edges are represented by an axial slice of the cutter. Helix angle of the cutter is ignored. The material removed during cutting is equal to the swept volume of the cutter in the workpiece. Effects of tool wear on the surface finish are ignored. Simulation system currently does not consider the effects of vibration. Surface finish is not affected by the initial nature of the flat surface. Surface error caused by deflection of the tool is ignored. These assumptions are used in the simulation model.

2.2 Simulation Model

Tool Model

A discrete axial slice is assumed to represent a cutting edge. A ball end mill is treated as a tool with semi-circular geometry while a flat end mill is considered as a flat cylindrical surface with fillet at

its end. Figure 1 shows the discrete axial slice of a 2-flute ball end mill, where R is the radius of the tool.

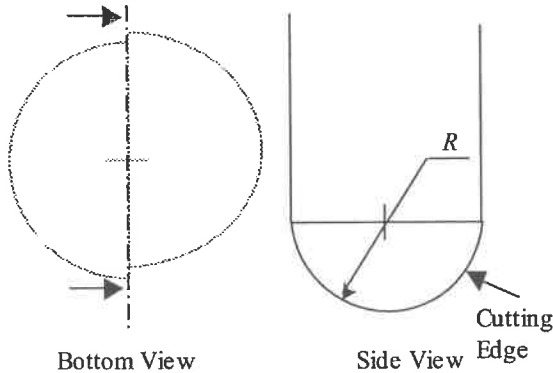


Figure 1 Tool model of a two-flute ball-end mill

Workpiece Model

Since the discrete model based simulation technique requires extensive calculations to determine the cut and uncut region, Z-map model based approach was chosen to represent the three-dimensional workpiece. In the Z-map model, the workpiece surface is divided into 2D grid (as shown in Figure 2). The z_{ij} coordinate of each grid point represents the height of the workpiece at that location, which changes with the cutting process. The z_{ij} coordinates are also associated with x_i and y_j coordinates of grid points and can be expressed as a function of $f(x_i, y_j)$.

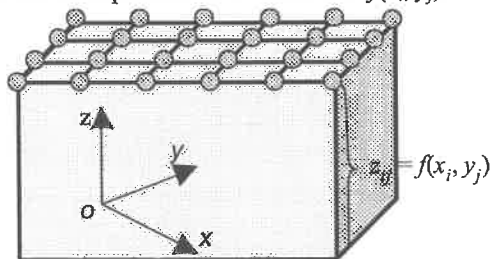


Figure 2 Z-map representation of workpiece

Tool Motion Modeling

The tool motion has two components: rotation and translation i.e., tool rotates with spindle and in the meantime translates with feed. Figure 3 shows the motion of the tool. Translation motion is assigned to the center point of the tool. The movement of the center of the tool will be controlled by the CNC program. In Figure 3, the 2-flute tool has two cutting edges 180° apart. The points on the cutter are mapped with respect to the center of the tool so that all the intersection calculations can be carried out based on the center position of the tool. The tool and workpiece intersection points are calculated based upon their interference. The workpiece material, which occupies the overlapped space between the the tool and workpiece, will be removed by machining. When the material is removed, the workpiece surface

is updated. The z coordinates of grid points on the new surface are updated accordingly. These new z coordinates, thus obtained after carrying out simulations, contains the information of the machined surface topography generated by the milling process.

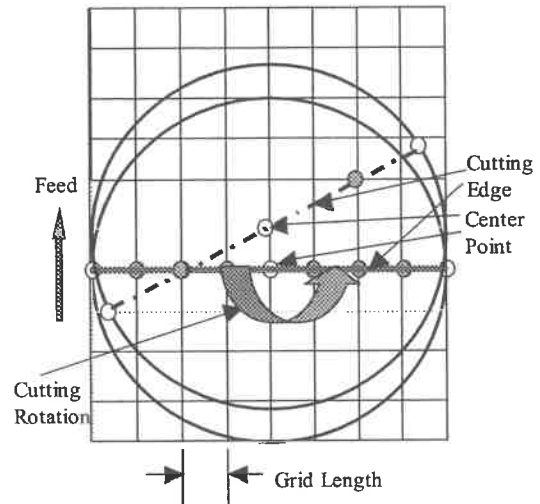


Figure 3 Tool discrete Motion

There are various parameters to represent the surface finish. In this research, spatial average surface roughness has been chosen as the parameter to represent the surface finish. The spatial average surface roughness calculation was based on the method suggested by Dong et. al. [5]. With VC++ (MFC) and OpenGL, a generalized simulation system was developed. Basic information such as type of tool, tool diameter, inclination angle and workpiece size and mesh size can be input to the simulation system through user interface. The simulation results are affected by the selection of mesh size and discrete incremental angle.

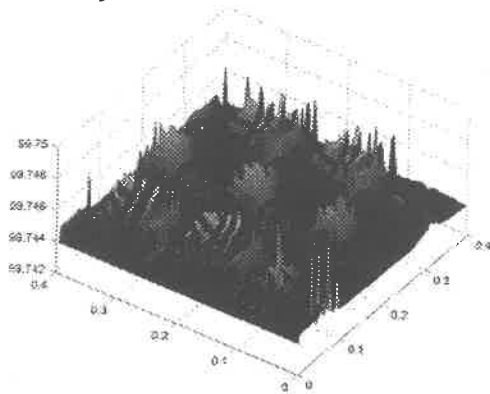
3 EXPERIMENT SETUP

The verification experiments have been conducted on Mori Seiki 5-axis (GV503/5Ax) and 3-axis (CV500) CNC machines. The average surface roughness of machined workpieces were measured by Zygo New View 5000. Ball end mills of $\Phi 6\text{mm}$ (2 flutes), $\Phi 10\text{mm}$ (2 flutes) and $\Phi 6.25\text{mm}$ (3 flutes) were used. Corresponding to the simulation, different feeds, pick feeds, spindle speeds and inclination angles were adopted to study the effect of these machining parameters. The CNC code was generated by using Feature CAM or a software program specifically written by the authors for various inclination angles. The workpiece materials used were aluminum and case hardened steel. Coolant was applied during machining

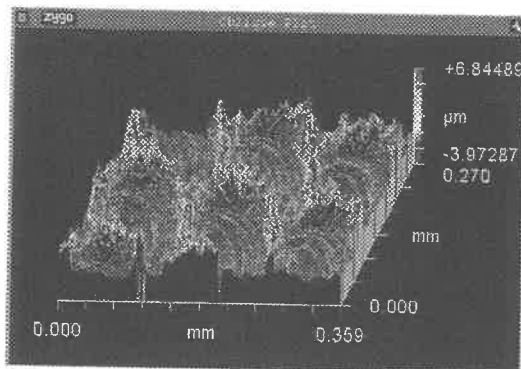
4 RESULTS AND DISCUSSIONS

4.1 Surface Topography

With the current model, the surface topography of milled workpiece can be simulated. Figure 4(a) shows the simulated surface topography. Compared to the measured result (Figure 4(b)), we can see that the current model has described the basic characteristics of milled surface. Due to the tool shape of ball end mill and cross feed, scallops are formed on workpiece surface during milling. Comparing these two figures, we can see that the scallop characteristics such as distribution on surface have been obtained by the current model. The calculated and measured heights largely agree with each other, except for a few variations on the measured surface. The difference may be due to material plastic deformation, which cannot be predicted by the model.



(a) Simulated surface Topography



(b) Measured surface topography

Figure 4 Surface topography

4.2 Surface Roughness

The preliminary tests were carried on a 3-axis machine tool, Mori Seiki CV500. The cutting tool used was 3-flute high speed steel ball end mill with a radius of 3.175 mm. The workpiece material was aluminum. Figure 5 shows the simulation and experimental results for surface roughness. The

simulation results show that surface roughness increases with the increase of feed rate, which agrees with the experimental results. For 3-axis machining, the difference between the predicted experimental results for surface roughness is pretty large. This may attribute to the vibration and spindle run-out in real machining, which were not considered in the simulation. The experiment has shown that higher cutting speed improves surface finish. Comparing the two simulation results under the same machining conditions but different discrete incremental angles, we can see that smaller incremental angle brings the simulated result closer to the experimental one.

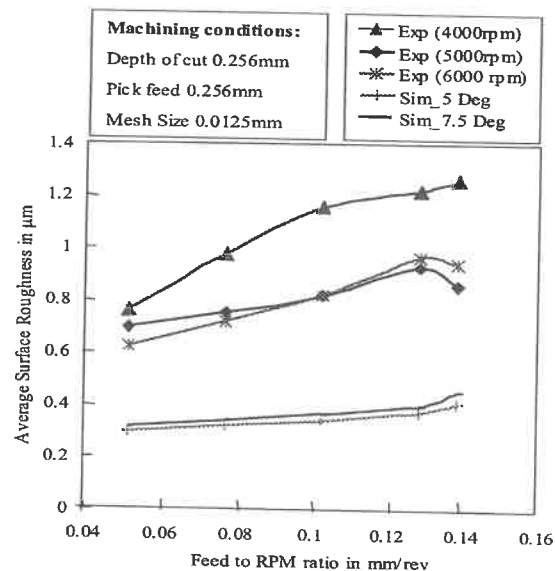


Figure 5 Simulation and experimental result for 3-axis machine

Further experiments were carried out on 5-axis machine tool, Mori Seiki GV503. These experiments were carried out by using 2-flute carbide coated ball end mill with a radius of 3mm. Spindle speed was maintained at 7000 rpm. The workpiece material was case hardened. In the simulation, 10-degree discrete incremental angle was used. The simulated and experimental results are compared in the Figure 6. The simulation has predicted the increase trend of surface roughness with the increase of feed rate. Compared to the above result by 3-axis machine, the current model predicts surface roughness by 5-axis machine better.

The simulation model was also applied to simulate the 5-axis machining, in which the cutting tool was inclined to workpiece surface cross feed directions. Figure 7 shows the effect of inclination angle in cross feed direction on surface roughness. The simulation result agrees with the average trend of the experimental result.

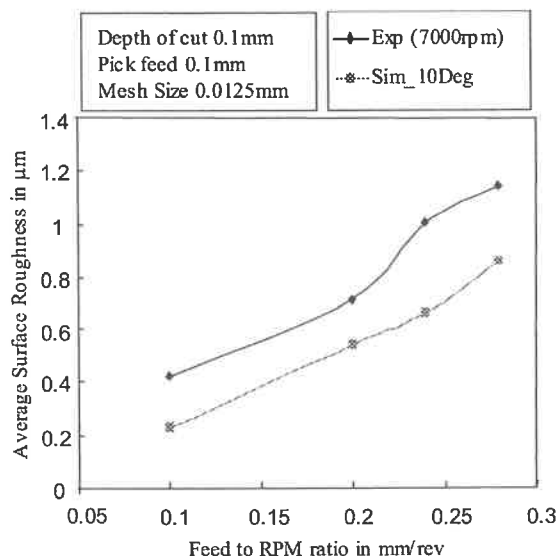


Figure 6 Simulation and experimental result for 3-axis machining on 5-axis machine

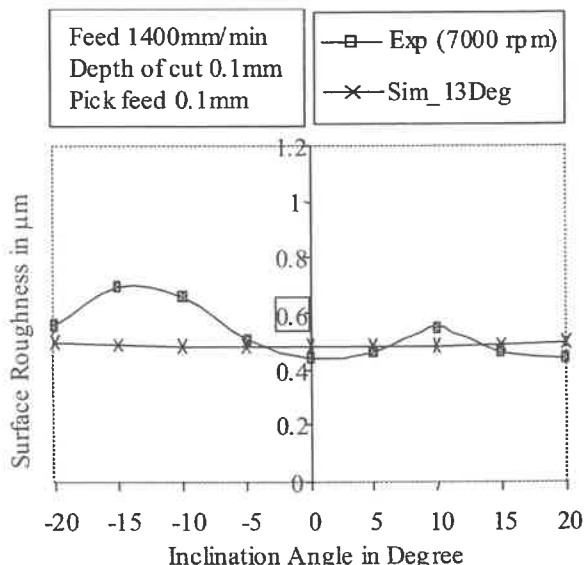


Figure 7 Surface roughness for 5-axis machining with inclination angle in cross feed direction

5 CONCLUSIONS

A simulation model based on Z-map method has been developed to calculate surface finish generated by 3- and 5-axis CNC machining. In order to verify the effectiveness of the model, machining experiments have been conducted under different conditions. This model has successfully predicted the surface topography, especially the scallop in CNC milling. The prediction for surface roughness largely agrees with the experimental results. The difference between the simulation and experiment may attribute to the effects of vibration, spindle run-out and workpiece material property in actual machining.

ACKNOWLEDGEMENT

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DEVELOPMENT OF AN INTERNET-BASED PLATFORM FOR HIGH-SPEED MILLING PROCESS PARAMETER SELECTION

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1.0 Introduction

This paper describes the initial development of an Internet-based application, 'Machinist Online', that will allow process planners to select high-speed milling parameters for maximized material removal rates in a science-based pre-process manner, rather than relying on experience. The primary mechanism for realizing this capability is dynamics prediction for the tool/holder/spindle/machine assembly using Receptance Coupling Substructure Analysis, or RCSA [1-4]. This method analytically predicts the assembly response by combining models and/or measurements of the individual components through empirically-obtained connection parameters. Currently, a primary impediment to full implementation of the academic research in high-speed machining, particularly linear and nonlinear chatter (or unstable machining) models, at the production level is the necessity of measuring each tool/holder/spindle/machine frequency response, typically by impact testing where an instrumented hammer is used to excite the structure and the response is recorded using an appropriate transducer, such as an accelerometer.

The chatter models, which can be used to select cutting conditions for both dramatic increases in material removal rates and improved part accuracy, require knowledge of the system frequency response as reflected at the tool point. Due to a potential lack of engineering support and sometimes limited knowledge of dynamic testing procedures, the frequency response measurements are rarely carried out, especially at Tier I and II manufacturing facilities that fabricate a large fraction of the total number of US discrete parts due to outsourcing from major automotive and aerospace manufacturers. Therefore, the well-established stability improvement technology (i.e., stability lobe diagrams, which separate stable and unstable cutting zones graphically as a function of chip width and spindle speed [e.g., 5-10]) afforded by high-speed machining is very often not applied. The result is reduced process efficiency and part quality and increased cost.

The development of the Internet-based parameter selection platform described here will require no frequency response measurements or specialized knowledge from the user. Rather, the user will select tool geometry and material, holder type, spindle model and manufacturer, and workpiece material from drop-down menus that access a local database. This information is passed to a MATLAB program using the MATLAB Web Server, where the RCSA frequency response prediction and corresponding stability lobe diagram calculation are completed. Using the stability lobe diagram, suggested operating conditions will be selected. The surface speed of the cutting tool is then compared to entries in TechSolve's web-based CUTDATA database [11], which contains entries for 3750 work materials and over 40 machining operations providing upwards of 90000 feed/speed recommendations. Due to temperature-driven chemical/diffusive wear, an upper bound on surface speed may be imposed on the stability lobe-based spindle speed. In this case, the selected spindle speed is revised to avoid excessive tool wear, while still maintaining a reasonable depth of cut. The final result is then passed back to the user. It is anticipated that the Internet tool will provide both the necessary predictive capability and a means of effective dissemination.

2.0 RCSA description

RCSA analytically predicts the tool point frequency response function (FRF) by coupling an experimental measurement of the holder/spindle substructure, or component, to an analytical or finite-element (FE) model of the tool through two empirical complex stiffness vectors, which include linear and rotational stiffness and viscous damping terms that characterize the non-rigid behavior of the connection between the holder and tool (e.g., thermal shrink fit, collet, or elastic deformation interference fit [12]). This technique is based on earlier receptance coupling work by Duncan [13], Bishop and Johnson [14], and, later, Ferreira and Ewins [15].

The model for the coupling between the holder/spindle and tool components is shown in Fig. 1. There are three translational and three rotational assembly coordinates identified, with spatial positions coincident with the coupling locations (coordinates X_2/Θ_2 on the tool and X_3/Θ_3 on the holder/spindle component) and the point of interest (coordinates X_1/Θ_1 at the free end of tool). The connection between X_2/Θ_2 and X_3/Θ_3 is composed of a linear spring, k_x , torsional spring, k_θ , linear viscous damper, c_x , and rotational viscous damper, c_θ . In order to determine the

assembly direct, or driving point, FRF at the tool point, $G_{11}(\omega) = X_1(\omega)/F_1(\omega)$, which is used as input to the selected process stability simulation, the following steps must be completed:

a) Use impact testing to measure the holder/spindle component (i.e., no tool inserted in holder) FRF, $H_{33} = X_3/F_3$, at the free end perpendicular to the spindle centerline. Typically, the measurement directions are selected to be coincident with the feed directions of the machine tool.

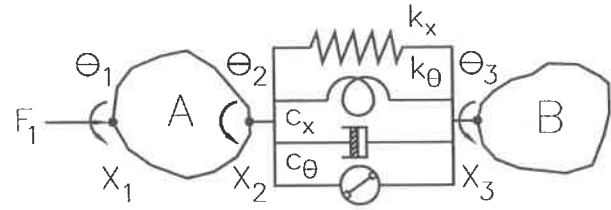


Figure 1: RCSA holder/spindle/tool model including connection parameters.

b) Develop an analytic model of the free-free tool using the closed form receptance terms, which capture both the rigid body and transverse vibration behavior of the tool, developed by Bishop and Johnson [14] (an FE model is also an option). Here, we have selected to treat the tool as an Euler-Bernoulli beam with a constant cross-section, which requires that an effective diameter, d_{eff} , be determined for calculation of the 2nd area moment of inertia, $I = \pi d_{eff}^4/64$. The effective diameter is based on the tool overhang length, L , total length, L_T , tool material density, ρ , shank diameter, d , and tool mass, M . See Eq. 1. Fundamentally, this equation calculates the diameter of a uniform cross-section beam with 1) a mass equal to the difference between the total tool mass and the mass of the tool shank inside the holder; and 2) a length equal to the overhang length of the tool, given the tool material density.

$$d_{eff} = \sqrt[4]{\frac{4M - \pi \rho d^2 (L_T - L)}{\pi \rho L}} \quad (1)$$

We have also added structural, or hysteretic, damping to the tool model by replacing Young's elastic modulus, E , for the tool material with the complex modulus, $E' = (1 + i\eta)E$, where η is the structural damping factor, a small dimensionless constant. This modifies the frequency-dependent term $\lambda = \left(\frac{\omega^2 m}{EI} \right)^{\frac{1}{4}}$ from Bishop and Johnson [14] to

be $\lambda' = \left(\frac{\omega^2 m}{(1 + i\eta)EI} \right)^{\frac{1}{4}} = \frac{\lambda}{(1 + i\eta)^{\frac{1}{4}}} \approx \lambda \left(1 - i \frac{\eta}{4} \right)$. To simplify notation, we drop the primes from E' and λ' in the

expressions shown in Eq. 2, which define the required free-free tool component receptance terms. In these expressions, different designations have been applied to the four receptance types found in our model, specifically,

$$H_{ij} = \frac{x_i}{f_j}, \quad L_{ij} = \frac{x_i}{m_j}, \quad N_{ij} = \frac{\theta_i}{f_j}, \quad \text{and} \quad P_{ij} = \frac{\theta_i}{m_j}.$$

$$\begin{aligned} \frac{x_1}{f_1}(\omega) = H_{11} &= \frac{-(\cos L\lambda \cdot \sinh L\lambda - \sin L\lambda \cdot \cosh L\lambda)}{\lambda^3 EI (\cos L\lambda \cdot \cosh L\lambda - 1)} & \frac{x_2}{f_2}(\omega) = H_{22} = H_{11} \\ \frac{x_2}{m_2}(\omega) = L_{22} &= \frac{\sin L\lambda \cdot \sinh L\lambda}{\lambda^2 EI (\cos L\lambda \cdot \cosh L\lambda - 1)} & \frac{\theta_2}{f_2}(\omega) = N_{22} = L_{22} \\ \frac{\theta_2}{m_2}(\omega) = P_{22} &= \frac{\cos L\lambda \cdot \sinh L\lambda + \sin L\lambda \cdot \cosh L\lambda}{\lambda EI (\cos L\lambda \cdot \cosh L\lambda - 1)} \\ \frac{x_1}{f_2}(\omega) = H_{12} &= \frac{\sin L\lambda - \sinh L\lambda}{\lambda^3 EI (\cos L\lambda \cdot \cosh L\lambda - 1)} & \frac{x_2}{f_1}(\omega) = H_{21} = H_{12} \\ \frac{x_1}{m_2}(\omega) = L_{12} &= \frac{\cos L\lambda - \cosh L\lambda}{\lambda^2 EI (\cos L\lambda \cdot \cosh L\lambda - 1)} & \frac{\theta_2}{f_1}(\omega) = N_{21} = L_{12} \end{aligned} \quad (2)$$

c) Measure the tool point response for the assembly in one direction at a known overhang. This data allows the determination of the connection parameters, k_x , k_θ , c_x , and c_θ , by nonlinear least squares best fit. This allows the model shown in Fig. 1 to be developed and analytic prediction of FRFs to be completed.

The desired tool point assembly receptance $G_{11}(\omega)$ is shown in Eq. 3, where we have substituted the complex, frequency dependent stiffness terms $K_x = k_x + i\omega c_x$ and $K_\theta = k_\theta + i\omega c_\theta$ to simplify notation; the full development of this equation is available in reference [4].

$$G_{11}(\omega) = \frac{X_1}{F_1} = H_{11} - \frac{H_{12}}{\det A} [K_x H_{21} (K_\theta (P_{33} + P_{22}) + 1) - K_\theta N_{21} K_x (L_{33} + L_{22})] \quad (3)$$

$$- \frac{L_{12}}{\det A} [K_\theta N_{21} (K_x (H_{33} + H_{22}) + 1) - K_x H_{21} K_\theta (N_{33} + N_{22})] \quad \text{where}$$

$$\det A = (K_x (H_{33} + H_{22}) + 1)(K_\theta (P_{33} + P_{22}) + 1) - K_x (L_{33} + L_{22}) K_\theta (N_{33} + N_{22})$$

3.0 Example experimental results

A tool/holder/spindle model was constructed using a two flute helical endmill, collet-type tool/holder connection, HSK-63A holder/spindle interface, and 20000 rpm spindle. The carbide endmill was 152.4 mm long with a 12.7 mm diameter shank and had a mass of 246.8 g. The flute length was 16 mm, the shank length was 65 mm, and the neck was relieved to a diameter of 11.1 mm. An allowable overhang range of 112.5 mm (8.9:1 length to diameter, or L:D, ratio) to 124.0 mm (9.8:1) was selected. The carbide density and modulus were taken to be $14.5 \times 10^3 \text{ kg/m}^3$ and $5.853 \times 10^{11} \text{ N/m}^2$, respectively [16]. In all measurements, the collet torque was set to the manufacturer-recommended value of 61 N-m (45 ft-lbf) using a torque wrench.

Following the algorithm described in Section 2.0, the holder was placed in the spindle and the direct FRF at the holder free end was recorded using impact testing in the vertical (y) and horizontal (x) directions for the horizontal spindle axis (z) machine configuration. Next, the tool receptances were calculated for a mid-range overhang length of 118.5 mm using Eqs. 1 and 2. A structural damping factor of 0.001 was assumed for the complex modulus calculation due to the difficulty in completing free-free FRF measurements on endmills. Finally, an x-direction tool point FRF was recorded for the tool/holder/spindle assembly and fit using Eq. 3 to determine the connection parameters and complete the RCSA model. The nonlinear least squares fit connection parameters are provided in Table 1.

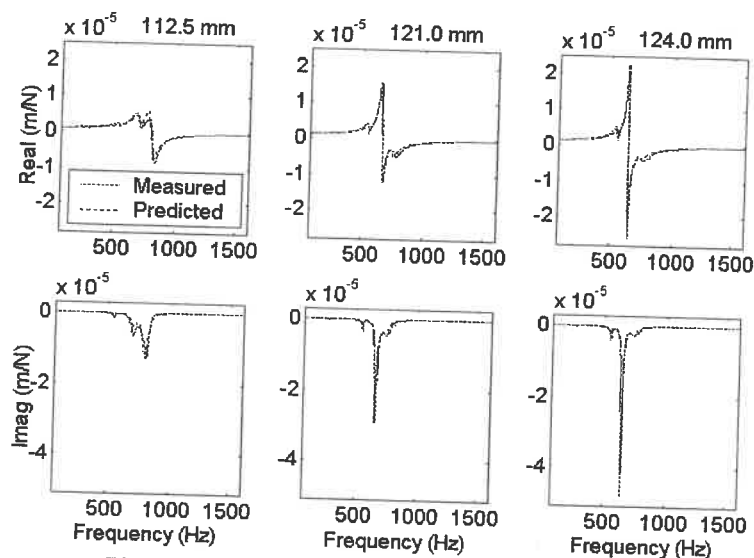


Figure 2. Assembly tool point FRF measurements and predictions for y-direction.

Table 1: Nonlinear least squares fit connection parameters.

k_x (N/m)	k_θ (N-m/rad)	c_x (N-s/m)	c_θ (N-m-s/rad)
6.8×10^7	2.7×10^6	3816	406

Predicted results for the y-direction using the fit parameters in Table 1 are shown in Fig. 2. The overhang lengths are the minimum and maximum values, 112.5 mm and 124.0 mm, respectively, and a near mid-range value of 121.0 mm. For the full range of measurements, good agreement was observed between the measured and predicted data in both the x and y- directions.

3.0 Web site description

As noted, the purpose of the 'Machinist Online' web site is to allow users without extensive knowledge of chatter theory or mechanical vibrations to take full advantage of the available improvements in process efficiency when using stability lobe diagrams. In the first generation of the web site, a measurement of the spindle/holder substructure, archived in a database, is coupled to the endmill via the experimentally-obtained connection

parameters, also contained in the local database, using RCSA. Once the tool point FRF is predicted, the corresponding stability lobes are generated using the analytic stability lobe algorithm developed by Tlustý [7] or Altintas and Budak [10]. These computations are carried out using MATLAB and the stability lobe diagram passed to the user using the MATLAB Web Server [17].

Inputs from the user are obtained either from nested drop-down menus or keyboard entry. These inputs include the machine/spindle/holder substructure (including the spindle specifications such as top speed and available power and torque), end mill material and geometry, workpiece material (specific cutting energy coefficients can be selected from the local database or input by the user), and radial immersion. Based on this information, recommendations for chip load and maximum surface speed are made using the web-based CUTDATA. The data passed back to the user includes the stability lobe diagram, corresponding spindle speed-dependent material removal rate (which may be power or torque limited in some instances), and recommended peripheral endmilling operating parameters, i.e., spindle speed and axial depth of cut.

The first release of 'Machinist Online' can be found at <http://highspeedmachining.mae.ufl.edu/>. The web site includes the predictive 'Parameter Selection' option, as well as a 'Parameter Simulation' module that will allow time-domain simulations of cutting force and deflections to be completed. It also includes a Frequently Asked Questions, or FAQ, section that provides introductory information on such topics as: 1) History of Chatter Research; 2) What is High-Speed Machining?; 3) What is Chatter?; 4) What is Impact Testing?; 5) What is a Stability Lobe Diagram?; 6) What is Tool Tuning?; and 7) What is RCSA? Descriptions of the University of Florida's Machine Tool Research Center and collaborating faculty and students are also provided.

The 'Machinist Online' will be continually updated to expand the machine, spindle, holder, connection parameters, and workpiece material databases. Additionally, future efforts will focus on a three-element assembly where the holder is considered separately from the spindle.

4.0 Acknowledgements

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Modeling and Investigation on a Jet Pipe Electrohydraulic Flow Control Servovalve

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Abstract The analyzed electrohydraulic servo valve is jet pipe type, is one of the mechatronics component used for precision flow control application. It consists of several precision and delicate components. For the analysis the jet pipe assembly of servovalve is identified and conducted a direct-solution steady-state dynamic analysis to study the response of the system for harmonic excitation. The flexure tube is one of the delicate and critical components in jet pipe assembly; the analysis was sighted around the flexure tube design and material property. The assembly was subjected for analysis to ascertain the response of the system for critical parameters like thickness of flexure tube and material for flexure tube. The system resonance frequency was observed for different flexure tube thickness and for different material properties of flexure tube.

Key words: flexure tube, resonance, jet pipe, steady state dynamics, flow control.

1. INTRODUCTION

Jet pipe electrohydraulic servovalve finds main application in actuating feedback control systems working on jet engine and fighter aircrafts. The primary flight control system for an aircraft is control-by-wire (CBW) and redundant digital processors are used to accept inputs from the cockpit controls and motion sensors and compute commands to the electrohydraulic actuators. Even though the leakage flow of the jet pipe amplifier is high compared to that of a flapper-type electrohydraulic servovalve, due to its capability of working in high-contaminated environment, it finds current application in fighter aircrafts. Jet pipe servo valves have one movable nozzle and two collector ports, from where fluid is ducted to the main valve spool. Pressure and flow in the collector ports are calculated from the principles of hydraulics, including a momentum balance. Thoma [1] has explained bond graph TUTSIM modelling approach, which is used to simulate the pressure in the receiver holes of the jet pipe servovalve. Dushes [2] has explained a pressure recovery in receiver holes of jet pipe servovalve, which is a function of jet pipe displacement relative to receiver plate. There are various parameters affecting the pressure recovery in the receiver holes. The major parameters are distance between the receiver holes (web thickness), jet pipe nozzle diameter, receiver hole diameters, jet pipe nozzle offset and jet pipe nozzle stand-off distance which are well explained [3]. Henry [4] has presented an analytical and experimental investigation of a jet pipe controlled electro pneumatic actuator for frequency response and time-domain force tracking. The intent of this paper is to present the response of the electrohydraulic servovalve component like jet pipe assembly for harmonic excitation.

2. JET PIPE SERVOVALVE

A schematic representation of a jet pipe electrohydraulic servovalve is shown in Figure 1. The servovalve consists of two main assemblies, a torque motor assembly representing the first stage and the valve assembly representing the second stage. In-between the first and the second stages, there is a mechanical feedback connected to the spool and jet pipe to stabilize the valve operation. The jet pipe serves to convert pressure energy of the fluid into the kinetic energy of a jet and directs this jet towards the receiver block where its kinetic energy is recovered in the form of pressure energy. The valve operates as follows:

- At first stage null, the jet is directed exactly between the two receivers, making the pressures on both sides of the spool equal. The force balance created by equal pressures in both end chambers holds the spool in a stationary

position.

- The first stage torque motor receives an electrical signal applied as current to the coils, and is converted into a mechanical torque on armature and jet pipe assembly.
- As the jet pipe and armature rotate around the pivot point of thin walled flexure tube, the fluid jet is directed to one of the two receiver holes in the receiver block, creating a higher pressure in the spool end chamber connected to that receiver. The differential pressure pushes the spool in opposite direction to the jet pipe displacement.
- As the spool starts moving, it pushes the feedback spring, creating the torque on the jet pipe to bring it back to null position. When the restoring torque due to spool movement equals the applied torque on the armature, the spool will stop in a particular position, until reversing the polarity of the applied current. This torque balance is said to be steady state operation of the servo valve. The resulting spool position opens a specified flow passage at the ports of the second stage of the valve.

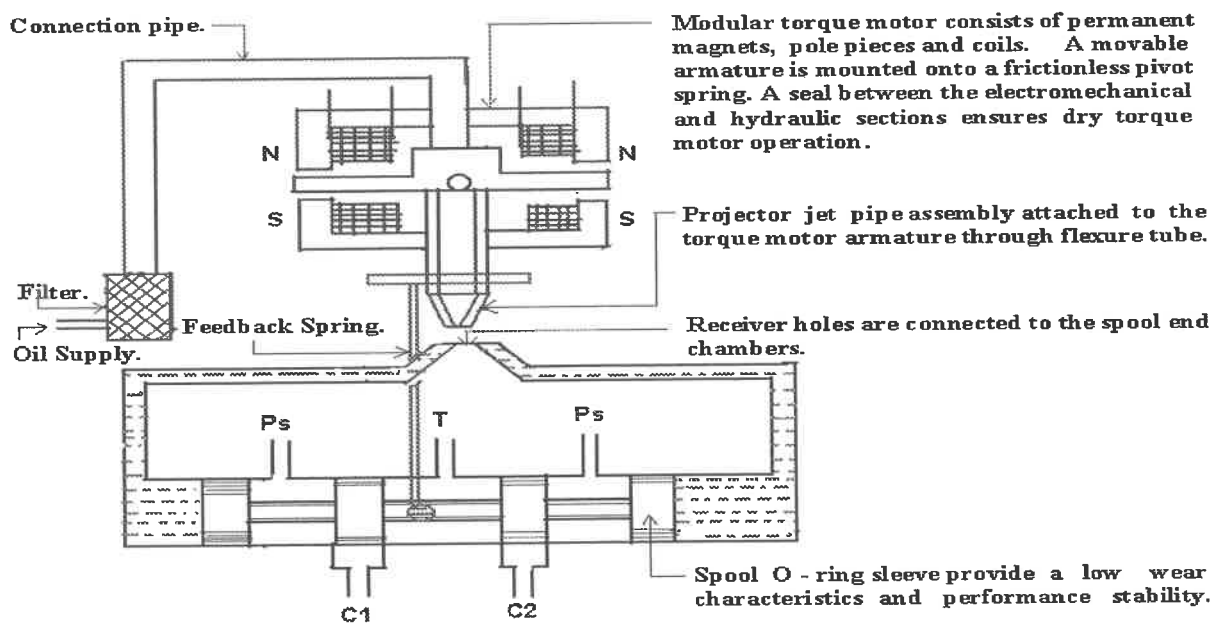


Fig.1 Schematic view of jet pipe servovalve.

3. DIRECT-SOLUTION STEADY -STATE DYNAMIC ANALYSIS

A direct-solution steady-state dynamic analysis is a linear perturbation procedure, used to calculate a system's linearized response to harmonic excitation. The solid model of the jet pipe servovalve is prerequisite for the analysis and was done using SDRC Ideas Ver. 7. With suitable boundary and loading condition, finite element analysis was carried out in Abaqus ver. 6.3. Fig.2 shows the solid and finite element model of jet pipe assembly, which consists of armature, armature bush, flexure tube, jet pipe, nozzle and oil supplying pipes. Depending upon the nature of operation, suitable elements are used in finite element with suitable material properties. The concentrated loads of 0.9414 N were applied on the armature, to create the torque on armature to rotate the jet pipe. These loads are sinusoidal with time over a user-specified range of frequencies. The jet pipe response was studied and was in Fig. 3, the jet pipe approaches the resonance frequency at 376.5 Hz with jet pipe maximum deflection as 1.6 E-01 m. These results were for flexure tube thickness of 45 micron and beryllium copper UNS 17300.

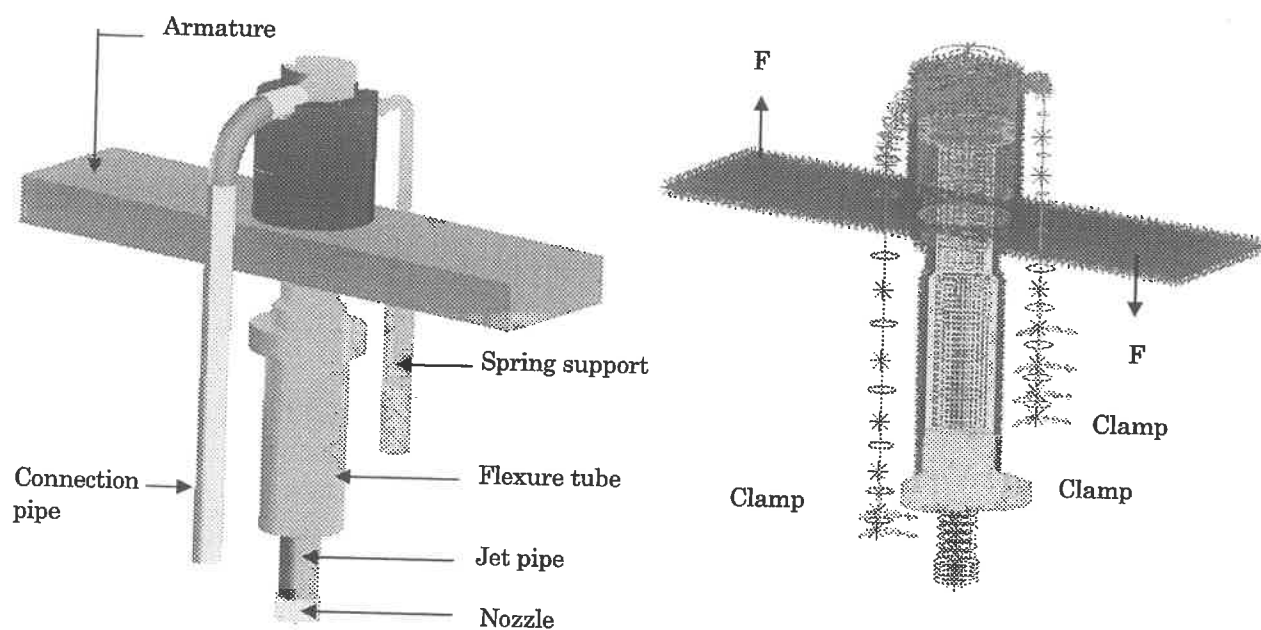


Fig.2 Solid and finite element model of jet pipe assembly

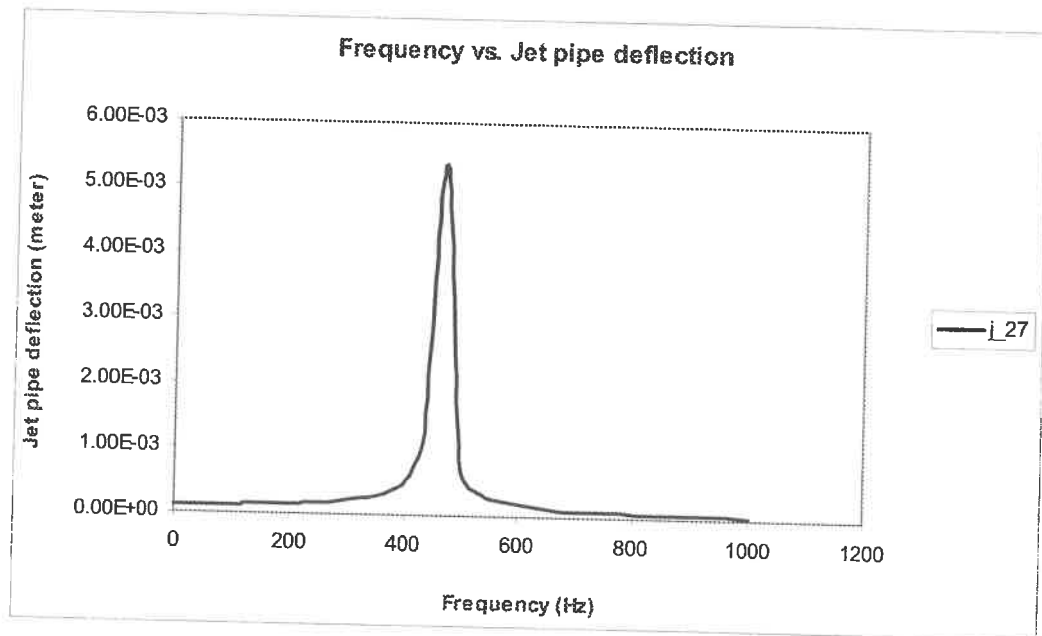


Fig.3 Jet pipe deflection vs. frequency

The analysis was extended to study the effect of flexure tube thickness and different material for flexure tube on natural frequency of the system and the obtained results were presented in Table 1. and Table 2. The natural frequency of the system increases with increasing thickness and material property of flexure tube was dominant parameter, which should be considered during the design of precision flexure tube.

Table 1: Frequency response of jet pipe assembly (Flexure tube material: UNS 17300)

t _{flex} (μm)	F _{arm} (N)	δ _{arm} (m)	δ _{jet} (m)	Resonance (Hz)
45	0.9414	9.389E-02	1.600E-01	376.5
50	0.9414	7.333E-03	1.245E-02	403.7
75	0.9414	8.555E-04	1.433E-03	533.7
100	0.9414	1.824E-03	2.999E-03	613.6

Table 2: Frequency response of jet pipe assembly (Flexure tube material : AISI 316)

t _{flex} (μm)	F _{arm} (N)	δ _{arm} (m)	δ _{jet} (m)	Resonance (Hz)
45	0.9414	3.164E-03	5.374E-03	464.2
50	0.9414	2.203E-03	3.732E-03	497.7
75	0.9414	6.403E-04	1.073E-03	657.9
100	0.9414	6.664E-04	1.092E-03	756.5

4. CONCLUSIONS

The flexure tube is one of the critical components in jet pipe assembly. The steady state dynamic analysis was conducted on jet pipe assembly and studied the response of the system. The critical parameters like thickness and material of flexure tube was studied for the frequency response.

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COMPUTATIONAL ACCURACY ANALYSIS OF A COORDINATE MEASURING MACHINE UNDER STATIC LOAD

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Abstract

This work presents a method of numeric simulation using finite element method (FEM) and geometrical analyses, to estimate the accuracy of a coordinate measuring machine (CMM) under static load, compared to its metrological behavior as a rigid body. A real CMM was CAD modeled and the deformation caused by a mass of 200 kg applied on its granite table was numerically analyzed by FEM. The deformed geometries of the machine guides were used in other computational analyses, now to estimate the measurement errors. The method is very practical to apply in any geometrical configuration of CMM, and useful of analyze the relation between the stiffness of the machine structure and its measurement uncertainty. Since it is not easy to calibrate a CMM with large masses on its table, a numerical simulation is valuable to estimate geometry deformations and avoid metrological problems.

Keywords: Coordinate Measuring Machine, Dimensional Metrology, Finite Element Method

1. INTRODUCTION

The operational and metrological performances of coordinate measuring machines have strongly improved in the last 10 years. Great technological evolution in all its components has improved the accuracy of the measurements and made the machines faster and more robust for industrial use. One of these evolutions was the implementation of software error correction, known as computer aided accuracy (CAA), taking the accuracy of the CMM beyond over its mechanical limitations. With CAA, mechanical errors are compensated mathematically by the measurement software according to error values obtained during calibration tests. In these calibration tests, the machine measures standards like gauge blocks, step gages, ball bars, etc., or its movement is compared against a laser interferometer. In all these processes, the machines are evaluated under some common conditions: the machine is calibrated without load over its table and the speed and acceleration of the machine axes are very small.

These conditions are not usual during measuring processes in industry. Many machines are used to measure parts with tens or hundreds of kg, and the measurement speed of machines axes used in the dimensional control of mass production is usually very high. This difference between calibration and measurement processes can cause problems on the software error correction, due to the static and dynamic stiffness of the machine. Deformations of the machine structure change the geometry and position of guides and axes, and the CAA works as if the machine was a rigid body.

This work presents a numeric simulation using finite element method (FEM) and geometrical analyses to estimate the loss of accuracy of a coordinate measuring machine when operating under static load, compared to its metrological behavior as a rigid body. The mechanical structure of a real CMM was CAD modeled and numerically analyzed using FEM. In these analyzes, a static load of 200 kg was imposed to the table of the machine and a deformation analysis revealed the deformed geometry. This deformed geometry of the machine was used in another set of simulations, now to analyze the accuracy of the machine to some measurements. The results of these simulations can show differences between the accuracy of the machine when operating with and without loads. Since it is not easy to calibrate a CMM with large masses on its table, a numerical simulation is valuable to estimate geometry deformations and avoid metrological problems. The method has been developed to estimate also errors caused by dynamic loads but, at the time of this paper, only static loads were simulated.

2. COMPUTATIONAL ANALYSIS OF A COORDINATE MEASURING MACHINE

To analyze the coordinate measuring machine under static load, the first step was to model its geometry on a CAD system. Based on a real CMM, the mechanical structure was CAD modeled. All parts, including table, guides and bearings were modeled based on real sizes of the machine and the assembly of individual parts was defined as realistic as possible. Based on this CAD model, a deformation analysis was implemented using finite element software. A load of 200 kg was imposed to the machine table, at the center of measuring volume of the CMM. The restraints, as well as, the materials (steel, granite and ceramic) of the machine parts were defined according to the real conditions. Meshes were created on the machine structure and the new geometry was calculated, resulting in a deformed configuration of the machine (figure 1).

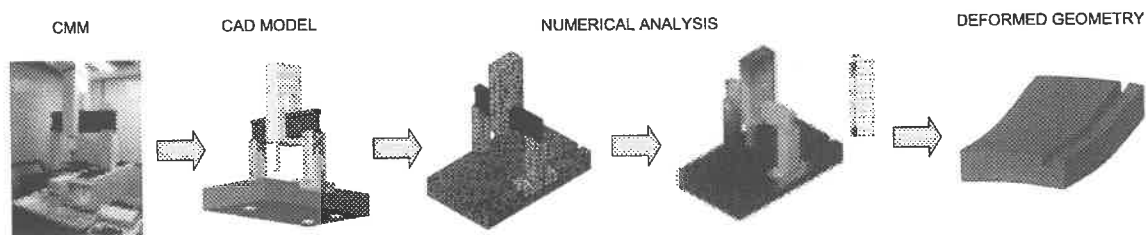


Figure 1 – Deformation analysis: from real CMM to deformed geometry

3. GEOMETRICAL SIMULATIONS OF MEASUREMENT ERRORS

The deformation analysis was performed with the machine bridge at several positions, in order to evaluate the path of the machine probe along the X axis (Figure 2). The position and displacements of some strategic points on the machine bridge and probe were monitored along X axis, in order to evaluate their path and angular errors. These results were used in another set of numerical analysis, now to determine the geometric errors of straightness (vertical and horizontal), Roll, Pitch and Yaw of the machine bridge along its path (X axis) over the guides on the machine table. Figure 3 shows the errors of straightness and the angular errors of roll, pitch and yaw of the machine bridge.

The maximum error of straightness was in the vertical direction (XTZ), since the mass load causes a bending on the machine table and guides of the bridge. The angular error of Pitch was the most critical, also because the bending of the table. But these errors became under acceptable values for this coordinate measuring machine. To some machines with structures of steel or aluminum, this influence could be more critical, and this analysis should be conducted to verify if the geometrical behavior is very affected by static loads.

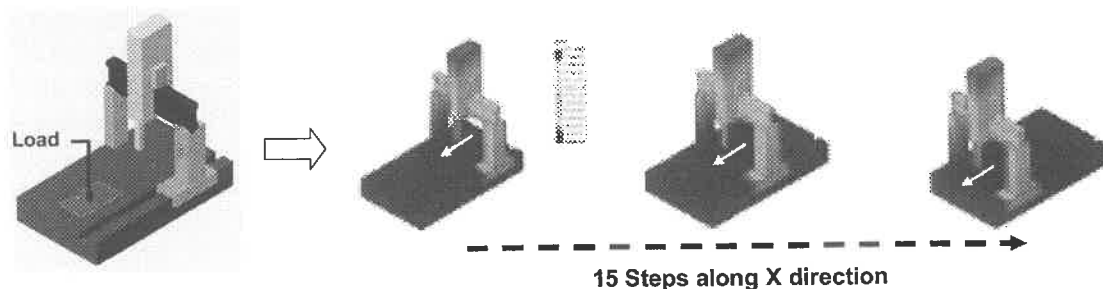


Figure 2 – Path of the bridge over a deformed granite table

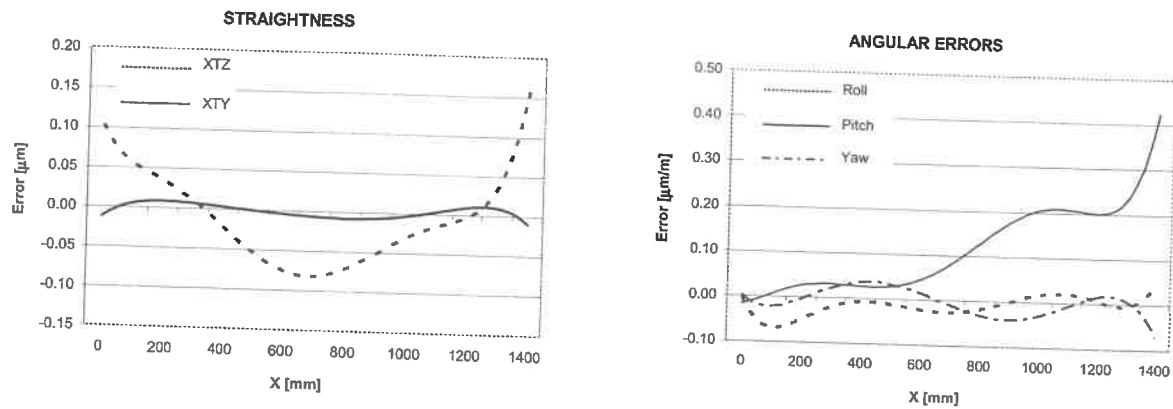


Figure 3 – Linear and Angular Errors of the machine bridge along its path

Using these geometrical errors, a geometrical simulation was performed to estimate the error that the machine would have to measure a length of 500 mm in some positions of the working volume. The measurements and results can be seen at figure 4. The errors of pitch and yaw were the main factors that influenced the results. It should be pointed out that this is the error caused only by the mass on the machine table and by the body forces of the machine itself.

From these results can be concluded that the stiffness of the machine table has a small influence over the results. Assuming that most part of the workpieces measured on this machine have less than 100 kg, it can be assumed that the static stiffness of the machine is suitable for its measurement task. Applying the same strategy it is possible also to make several other simulations of errors for other measurement tasks (diameter, squareness, straightness, etc.). The position of the workpieces in the measuring volume must be indexed for the analysis.

More than analyze this machine, this paper has the objective to present a methodology to verify numerically the metrological behavior of coordinate measuring machines submitted to static loads of heavy workpieces.

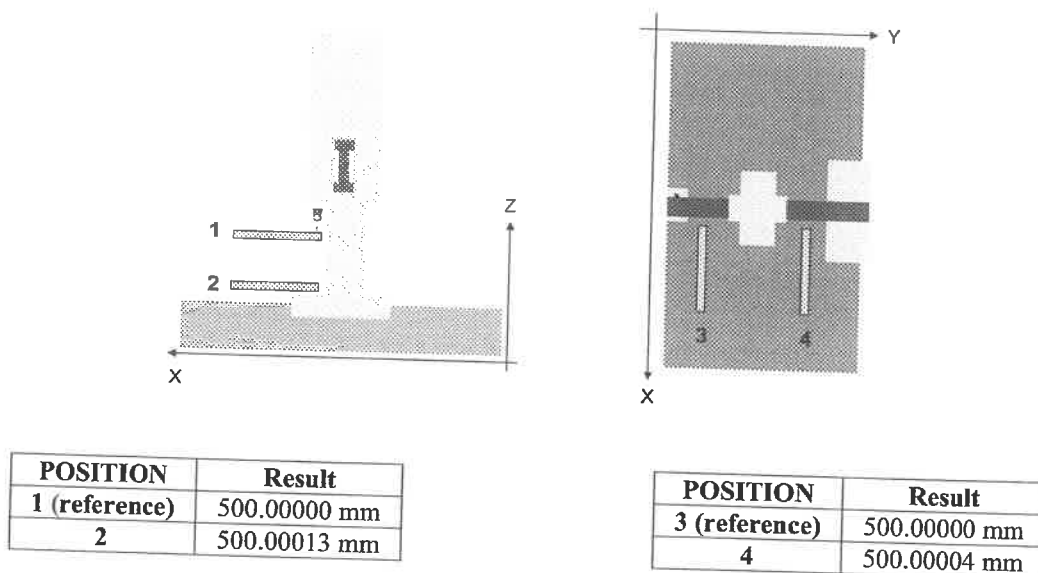


Figure 4 – Different measurement results at different positions

4. CONCLUSIONS

This work presented the use of numerical methods to evaluate the metrological behavior of a coordinate measuring machine under static load. Applying simulation using finite element method (FEM) and geometrical analyses, it is estimated the accuracy of a coordinate measuring machine (CMM) under heavy static load, compared to its metrological behavior as a rigid body.

An evaluation was performed over a coordinate measuring machine with granite table. The machine was CAD modeled with the materials and restraints defined according to real conditions. A static load of 200 kg was applied on the machine table and the deformed geometry was used to estimate geometric deviations of the bridge along its path. Calculating deformed geometries for different positions of the machine probe along the working volume it is possible to estimate errors for different measurement tasks.

The simulation is practical of implementation and the results are very useful to estimate the metrological behavior of the CMM, mainly in situations where the workpieces have tens or hundreds of kg. The machine analyzed in this paper has a very strong mechanical structure, not more found in modern machines, which have been optimized to be lighter and faster.

The application of numerical analysis to evaluate geometric error of precision machine can be a powerful tool to verify the influence of some factor that affects its accuracy. Situations that would be difficult to verify experimentally can be estimated with good reliability. In this paper, a mass of 200 kg was loaded on the machine table. Other simulations of the machine behavior under dynamic loads and thermal loads have been tried and the results will be discussed on next papers.

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Adaptive Feedforward Cancellation Viewed from an Oscillator Amplitude Control Perspective

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Abstract

This paper summarizes the design and implementation of Adaptive Feedforward Cancellation (AFC) on a fast tool servo (FTS) for use in high-precision diamond turning applications. Experimental results are presented and compared to those of a conventional integral controller. A method of viewing AFC from an Oscillator Amplitude Control (OAC) perspective is also discussed, which provides additional use of classical control techniques to determine the convergence and error properties of AFC resonator systems.

Keywords: Adaptive Feedforward Cancellation, Fast Tool Servo, Oscillator Amplitude Control.

Introduction

In diamond turning applications, fast tool servos commonly follow near-periodic trajectories, since the tool motion is keyed to the fundamental spindle rotation frequency. The FTS axis can develop significant following errors, since conventional feedback loops (*i.e.*, PID and lead-lag) only provide a finite controller gain at non-zero frequencies, even when command pre-shifting feedforward is included [1]. The error signal primarily consists of a summation of sinusoids of known frequencies and unknown Fourier coefficients. In the literature, a number of methods exist for the rejection of this periodic error. In this paper, we focus primarily on the applicability of AFC for reducing/eliminating the steady-state tracking errors of fast tool servos in high-precision diamond turning applications.

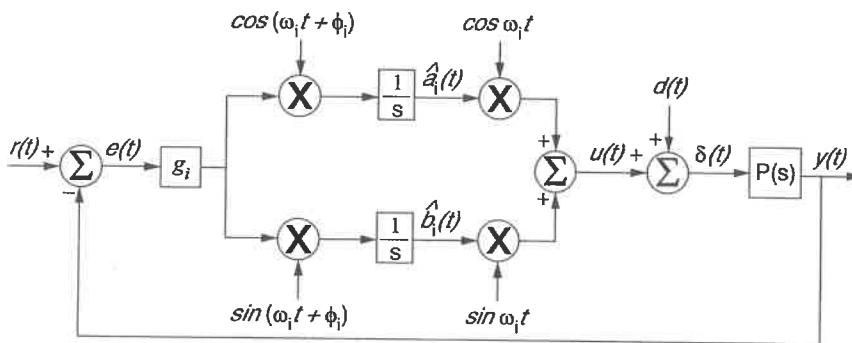


Figure 1: Single resonator AFC closed-loop block diagram.

Figure 1 illustrates the loop configuration for a single resonator AFC controller, where $P(s)$ is the transfer function of the plant being controlled. This system is designed to provide zero steady-state error at frequency ω_i . The internal dynamics of the AFC algorithm are linear time-varying (LTV) but in the literature it is shown that the input-output relationship from $e(t)$ to $u(t)$ is equivalent to a linear time-invariant (LTI) system [1]. The equivalent continuous-time transfer function is given by

$$C_i(s) = g_i \left[\frac{s \cos \phi_i + \omega_i \sin \phi_i}{s^2 + \omega_i^2} \right], \quad (1)$$

where g_i is a proportional gain and ϕ_i is a phase advance parameter that can be chosen to improve the closed-loop system's robustness.

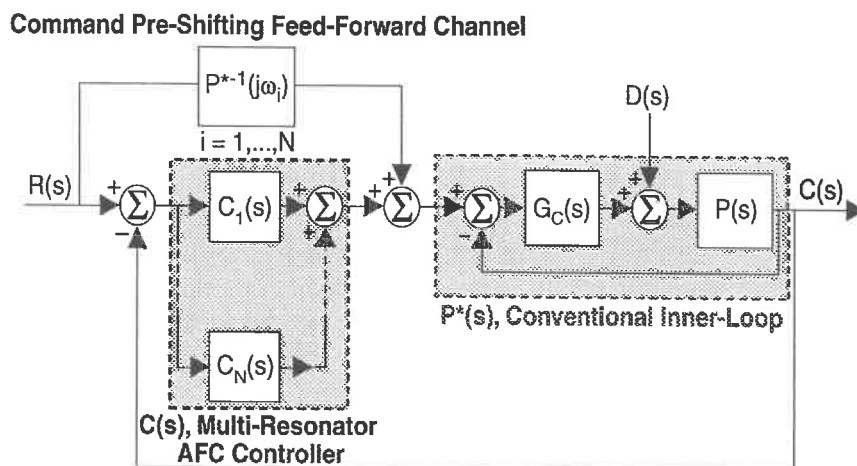


Figure 2: AFC feedback loop configuration used with the Variform FTS.

In order to be able to provide zero steady-state tracking error to multiple harmonics, several AFC resonators can be placed in parallel to form a multiple resonator AFC system. Byl *et al* [1] have developed a loop-shaping approach to designing these multi-resonator AFC systems. Their complete design, as shown in Figure 2, includes a conventional inner-loop controller $G_c(s)$, command pre-shifting feedforward channel $P^{*-1}(j\omega_i)$, and multiple resonator AFC controller $C(s)$.

Experimental Hardware

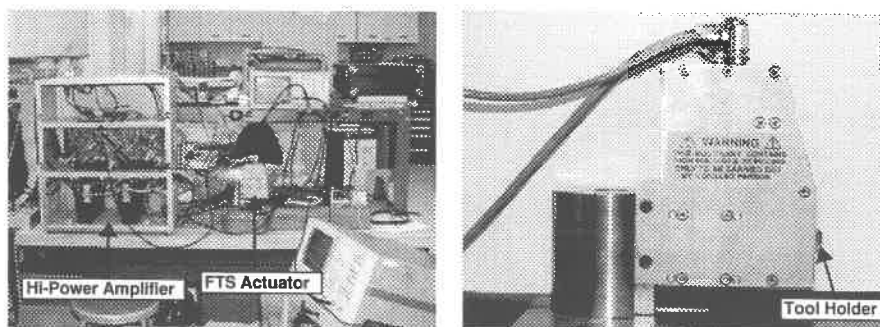


Figure 3: (left): Variform FTS and experimental hardware. (right): Close-up of the Variform FTS actuator.

We used the previously mentioned loop-shaping approach and the controller configuration in Figure 2 to implement a ten resonator AFC system on a commercially available FTS. This system, the Variform FTS, was designed and manufactured by Kinetic Ceramics [2] and is shown in Figure 3. This particular FTS actuator consists of a tee-lever mechanism actuated by a double piezoelectric (PZ) stack arrangement coupled to two H-plate flexures. A high-power amplifier, with an on-board digital feedback controller, drives the PZ stacks differentially to a maximum of ± 400 V, which provides about a 200 Hz 0 dB crossover frequency with a total displacement of up to ± 250 μm ($\pm 0.010''$).

The Variform FTS on-board controller takes advantage of an inner charge loop to minimize the PZ hysteresis curve along with an outer position feedback loop. During our FTS experiments, we utilized a PC-based digital control system to apply our AFC algorithms. Thus, we bypassed the on-board controller, while keeping the inner charge loop enabled, and used the PC-based digital controller to perform all of our closed-loop experiments.

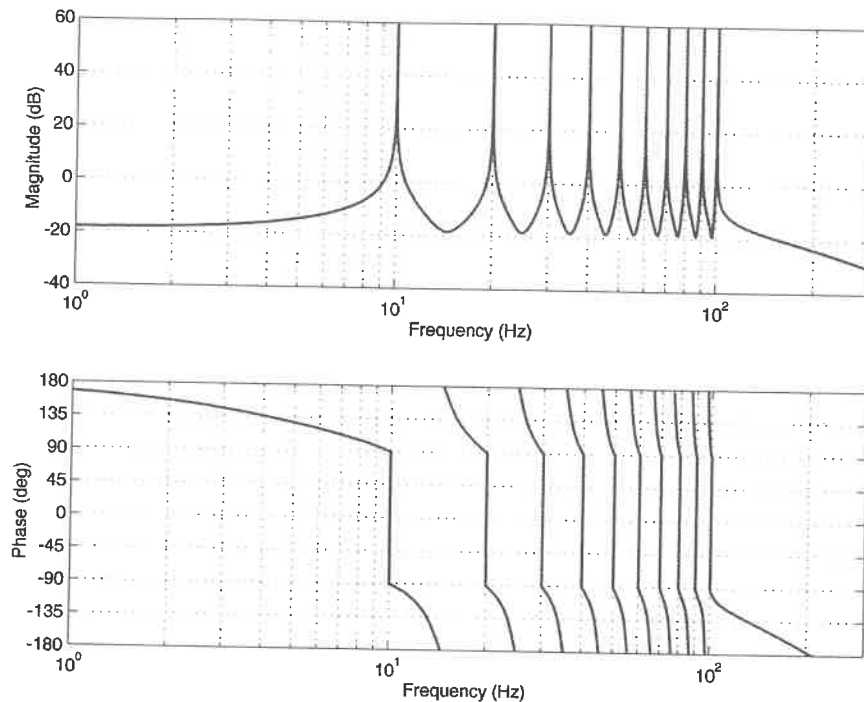


Figure 4: Calculated Variform FTS negative of the loop transmission frequency response with a ten resonator AFC and conventional inner-loop integral controller.

Experimental Controller

Figure 4 illustrates the calculated negative of the loop transmission frequency response for the Variform FTS with a multi-resonator AFC controller and conventional inner-loop integral compensator. The design of these loops is described in detail in [4]. This particular system is designed to provide zero steady-state error to a signal with up to ten harmonics at frequencies of

$$\omega_i = 10 \text{ Hz}, 20 \text{ Hz}, \dots, \text{ and } 100 \text{ Hz}. \quad (2)$$

Results

Figure 5 shows the experimental closed-loop error signal during an air-cutting experiment, which includes the previously mentioned ten harmonics. Without AFC, the peak-to-peak system error amounts to approximately 15% of the reference signal, while the peak-to-peak system error with AFC control reduces to about 0.5%. Note that the error signal with AFC control is dominated by the noise of the feedback sensor and there appears to be no apparent signal left that is correlated to the input reference signal.

Oscillator Amplitude Control

Roberge [3] showed that the amplitude of a sinusoidal oscillator can be stabilized by using an auxiliary feedback loop. He refers to this approach as an oscillator amplitude control system and states that if the bandwidth of this loop is much lower than the frequency of oscillation, then we can analyze the amplitude dynamics alone and ignore the sinusoidal portion of the loop. A detailed discussion of Roberge's oscillator amplitude control system is located in [4], which also develops a method of viewing AFC from an OAC perspective.

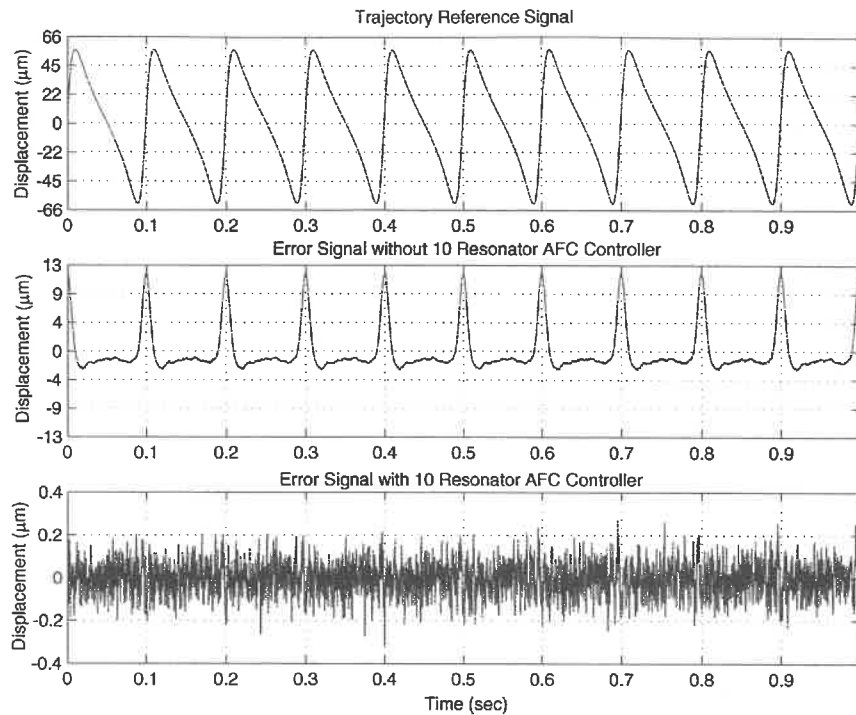


Figure 5: Comparison of the experimental closed-loop error signal with and without AFC Control.

Viewing AFC from an OAC perspective provides an approximate measure of the amplitude dynamics of an AFC closed-loop system. This perspective also allows the additional use of classical control techniques to determine an AFC loop's stability, convergence, and error properties. In [4], Cattell uses an averaging analysis to simplify a properly tuned single resonator AFC system into two single-input single-output amplitude control loops. With these loops de-coupled, the performance of any combination of AFC resonators can be approximated by the superposition of N OAC loops.

Conclusions

The results of these AFC experiments illustrate a dramatic improvement in the Variform FTS's steady-state tracking performance to constant amplitude periodic signals. However, determination of the AFC closed-loop convergence and error properties to changes in the reference/disturbance amplitude is not an obvious output of these analyses. Thus, we can simplify any AFC resonator system into the sum of N OAC loops and approximate the performance from the amplitude dynamics alone, independent of the detailed time variation of the AFC output signal.

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Development of an electro-dynamic planar motor

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Abstract

This poster describes an electro-dynamic planar motor, developed for use in equipment for accurate positioning of substrates. It provides a way out of the design size and complexity for a stacked axes drive and provides additional benefits for use in vacuum equipment. Multiple levels of electromagnetic analysis are used to optimize the design and identify the required control algorithms.

Introduction

Conventionally, combined X-Y movements in positioning equipment are realized by stacking of linear motors. This solution offers limited accuracy: increasing mass in combination with finite mechanical stiffness limits the obtainable control bandwidth and thus disturbance rejection and accuracy. Application of a planar actuator layout provides a way out. The layout can be combined with the use of electro-dynamic actuators, making use of their superior dynamic performance.

On the other hand, a levitated electro-dynamic motor provides distinct advantages for use in vacuum equipment. It does not have contamination sources like greased roller bearings, nor does it require the extensive mechanics that are connected to scavenging air bearings.

Principle of operation

The basic actuation principle is electro-dynamic. The main driving force is generated efficiently (in terms of specific force per motor volume) due to the special orientation of the coils relative to the magnet matrix. This orientation allows for the total planar area to be filled with magnetic poles. See Figure 1.

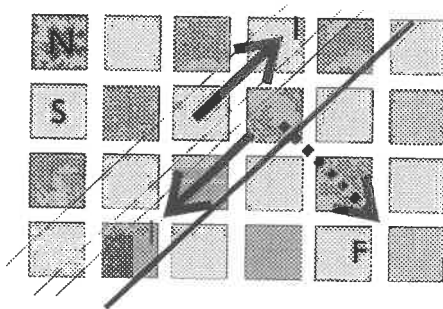


Figure 1 Force generation

The electro-dynamic force generation is made even more efficient by adding so called Halbach magnets [1] in the magnet matrix. Three coils combine to a forcer unit, which can be seen as a three phase brushless motor. By introducing an extra phase angle in the electronic commutation, one forcer unit is capable of generating forces in two degrees of freedom, (x and z , see Figure 3 and Figure 2). On top of that, it will generate a torque about the remaining degree of freedom that is referred to as the pitch effect. This R_y torque component will produce x and z -forces and will thus introduce disturbances in the x and z position control loops. Using two slightly displaced coil

units, carrying the same current, and choosing suitable coil geometry, minimizes disturbances due to this pitch effect [3].

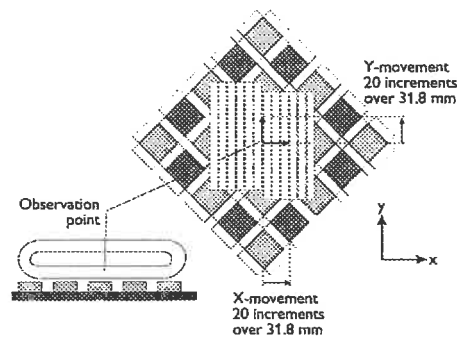


Figure 2 Planar force unit

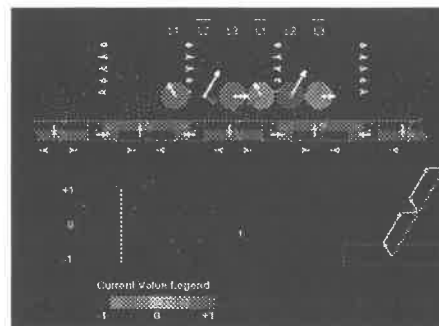


Figure 3 Force generation with three phase current

The complete planar motor forcer consists of four units, and is capable of generating forces in 6 degrees of freedom. In the plane, the main x - and y -forces are generated. These will be used to actuate the major displacements of a substrate carrier. The z -force and torques are less powerful and will be used to stabilize the remaining degrees of freedom. See Figure 4.

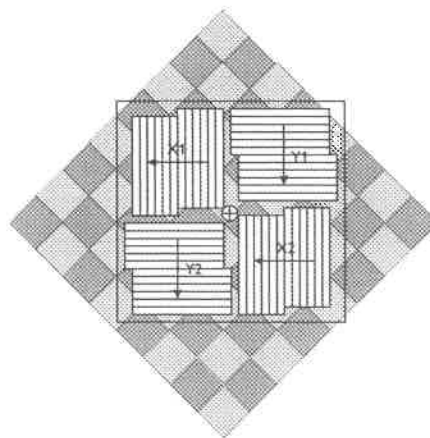


Figure 4 Total planar motor forcer layout

To address the 6 degree of freedom MIMO control problem, a static decoupling control approach is used together with static velocity and acceleration feedforwards. This approach works well if a rigid body behavior for the planar

motor mover can be assumed. In a prototype application, the bandwidth-limiting dynamics were found to be at 300 Hz, justifying this approach.

Magnetic analysis

Optimization of the electromagnetic geometry has been achieved using both analytical as well as finite element analysis techniques. For both of these, 2D as well as 3D analyses were performed. The geometry of the planar motor was optimized using three-dimensional field analyses to obtain minimal cross-talk in the various degrees of freedom and the most advanced, patented magnetic field topology (Halbach array). Earlier investments in two- and three-dimensional field analysis, analytical as well as numerical meant less work when it came to developing a control solution for this motor.

Application

Depending on the required accuracy, a number of solutions are possible to obtain position information, including laser interferometers, capacitive sensors, hall sensors or encoders located in mechanical motor arms. The first prototype used encoder sensors, mounted on an external mechanical gantry (see Figure 5).

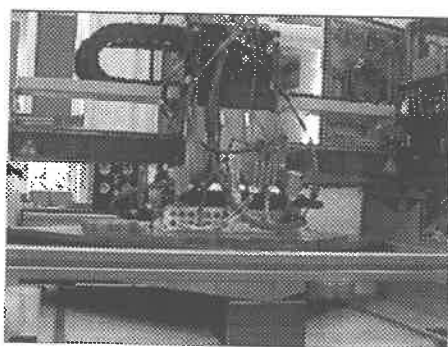


Figure 5 Prototype planar motor

The magnetic field is realized by placing magnets on a base plate and coils above the base plate. The X and Y forces are controlled through the use of defined algorithms. Extensive efforts aimed at developing a new type of coil cooling have resulted in a combination of water cooling and technical ceramics.

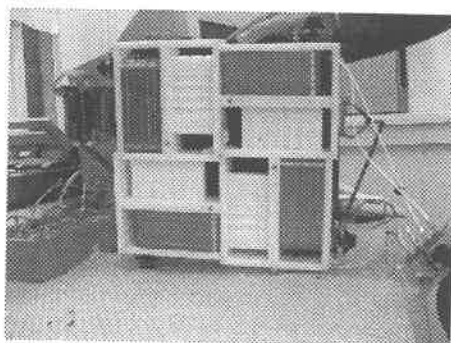


Figure 6 Forcer with coils (only one per forcer unit)

Conclusion

The planar motor is still in a developmental stage. Many groups within Philips CFT are contributing their various expertises to this innovative and very promising planar motor concept. Ultimately the motor will feature an accuracy of 10 μm , a speed of 1 m/s and an acceleration of 10 m/s^2 .

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Model-based Control Input Compensation for Parallel Kinematic Machine Tools

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Abstract

This paper presents a new model-based approach for control input compensation leading to better performing machine tools with nonlinear behavior. An optimization algorithm has been developed and implemented for the parallel kinematic machine tool (PKM) *Hexaglide*, developed at the IWF in Zurich. With the developed algorithm the contour feed along a given path is maximized offline according to the limited maximum actuator speed and force leading to minimization of execution time. In addition to the capabilities of similar algorithms the presented solution supports automatic optimization of an entire NC program. Some basic considerations are presented for the dynamics of PKMs. The optimization algorithm is also described. Results of optimized trajectories are shown. Their execution time is compared with traditionally created trajectories using constant limits.

Keywords: Optimization, Control, CNC.

1 Introduction

The treatment of control input is called *Control Input Compensation (CIC)* (see [1]). Several areas and methods for CIC have been examined in the past, e.g. time or frequency-based filtering of the controller inputs; geometric modulation of the path within certain tolerances, e.g. prevention of corners and discontinuous curvature by smoothing the path [2]; contour feed control along a given path [3]/[1]. In this paper, *contour feed control* along a given path is regarded. Hereby, appropriate criteria have to be kept. Typical criteria are *maximum actuator speed* $\dot{q}_{max}$ and *maximum actuator force* τ_{max} . Traditional algorithms introduce velocity profiles (e.g. trapezoidal or $\sin^2$ profiles) where discontinuities in the feed at corners or changing feed rate occur. In general, the machine tool is assumed to behave linearly. Mainly in the field of PKMs - built for high dynamic applications - special attention has to be paid to CIC. The sig-

nificant nonlinearities of a PKM lead to strong dependencies of the dynamic capabilities on the actual position and orientation of the tool. Therefore traditional algorithms for the CIC described above clearly fail to fully utilize the capabilities of PKMs.

1.1 Some examples of nonlinear behavior

The following figures show examples of nonlinear behavior. The first example illustrates *direction dependency* of the contour feed limit, whereas the second example illustrates *position dependency* of acceleration limits.

Figure 2 shows maximum contour feed dependency on its direction in the Y-Z plane for the PKM *Hexaglide* (see figure 1). The Hexaglide is a 6-DOF milling PKM with constant strut length and moving base-points ("glide"-principle). Its equations of motion are significantly nonlinear.

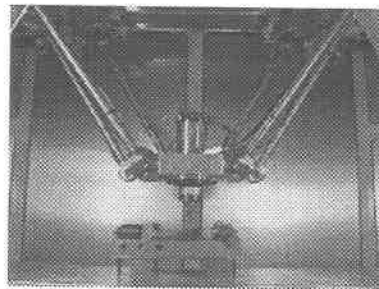


Figure 1: PKM Hexaglide, developed at the IWF Zurich. In the middle of the figure, the spindle is shown. On the top of the figure, the actuators (linear motors) are shown.

Limits regarding maximum actuator speed and force are plotted. The resulting area of allowed feed is compared with a square area defined by conventional CIC (hatched area). Thereby, constant feed limits for the Y and Z direction are used (see also section 5). As it can be seen, the area defined by conventional CIC is considerably smaller than the model-based one.

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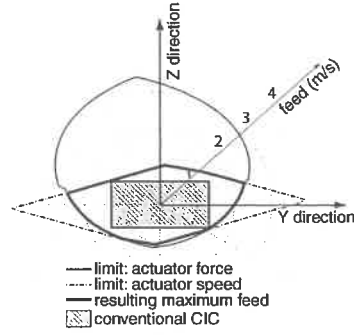


Figure 2: Polar plot for direction dependent maximum contour feed in the Y-Z plane. Computed for the PKM *Hexaglide*. Tool center point position is the approximate working volume center: X=0, Y=0, Z=-1 m.

The position dependent maximum acceleration in +Y direction from standstill is shown in figure 3. Maximum acceleration varies from near 0 to 25m/s^2 in the working volume. Conventional CIC would have to reduce the acceleration in +Y direction to a very low limit that is valid for the whole working volume. Therefore the machine tool would move with acceleration far from possible in most of the working volume.

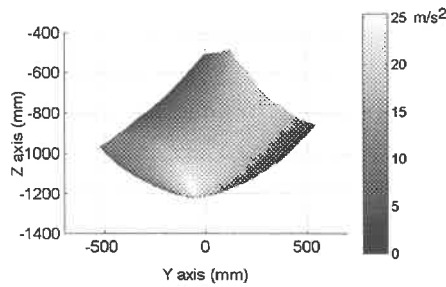


Figure 3: Maximum acceleration in +Y direction in the working volume for the Y-Z plane (X=0). Computed for the PKM *Hexaglide*.

2 Task formulation

2.1 Tool path formulation

Paths for machine tools are typically given by *NC programs*. Thereby, geometric commands define the nominal contour. Process requirements define the feed along the path during machining, which has to be expressed in the NC program by programming of

the *feed rate*. In contrary, movement during *rapid traverse* should be executed as fast as possible (e.g. tool or workpiece change).

For each geometric command i (block index i) of an NC program, the path is given analytically (e.g. a circular arc). The 6-tuple $\mathbf{x}$ containing the position and orientation of the tool in the workpiece coordinate system is given as a function of a path parameter s (e.g. arc length of the path):

$$\mathbf{x}^i = \mathbf{x}^i(s), \quad s_0^i \leq s \leq s_1^i \quad (1)$$

1st and 2nd time derivatives:

$$\dot{\mathbf{x}}^i(s(t)) = (\mathbf{x}^i)'(s) \dot{s} \quad (2)$$

$$\ddot{\mathbf{x}}^i(s(t)) = (\mathbf{x}^i)''(s) \dot{s}^2 + (\mathbf{x}^i)'(s) \ddot{s} \quad (3)$$

2.2 Machine model

According to [4], the equations of motion in the form of the inverse dynamic problem of general mechanical multi-body systems can be expressed as the *configuration space equation*

$$\begin{aligned} \tau &= \mathbf{M}(\mathbf{x})\ddot{\mathbf{x}} + \mathbf{B}_1(\mathbf{x})\kappa_1 + \mathbf{B}_2(\mathbf{x})\kappa_2 \\ &\quad + \mathbf{C}(\mathbf{x})\dot{\mathbf{x}} + \mathbf{g}(\mathbf{x}), \\ \kappa_1 &:= (\dot{x}_1\dot{x}_2 \quad \dot{x}_1\dot{x}_3 \quad \dots \quad \dot{x}_{n-1}\dot{x}_n)^T, \\ \kappa_2 &:= (\dot{x}_1^2 \quad \dot{x}_2^2 \quad \dots \quad \dot{x}_n^2)^T. \end{aligned} \quad (4)$$

Combination of equations 1-4 results in

$$\tau = \mathbf{m}(s^i)\ddot{s}^i + \mathbf{b}(s^i)(\dot{s}^i)^2 + \mathbf{c}(s^i)\dot{s}^i + \mathbf{g}(s^i) \quad (5)$$

for NC block i .

With equation 5, the equation of motion ("along the path") is expressed in terms of the path parameter s^i and its time derivatives. This accords to a *reduction* of the information to the *effectively needed* information to solve the problem of contour feed control. This fact is expressed in the reduction of the system matrices $\mathbf{M}$, $\mathbf{B}$, etc. in equation 4 to vectors $\mathbf{b}$, $\mathbf{c}$, etc in equation 5.

In equation 5, most of the relevant physical effects on multi-body-systems are modelled. But the important part of Coulomb friction in the actuators is still neglected (see [5]). Therefore, equation 5 has been completed by the following term representing Coulomb friction:

$$\begin{aligned} \tau_c &= \mathbf{w}(s), \\ \mathbf{w}(s) &:= \begin{cases} 0 & \dot{s} = 0, \\ \text{sign}\{\mathbf{J}^{-1}(\mathbf{x})\mathbf{x}'(s)\} \mathbf{W}_c & \text{otherwise.} \end{cases} \quad (6) \\ \mathbf{W}_c &:= \text{diag}\{w_{c_1} \quad \dots \quad w_{c_n}\} \end{aligned}$$

whereby $\mathbf{W}_c$ is a diagonal matrix, containing the Coulomb friction forces and $\mathbf{J}$ is the Jacobian.

2.3 Optimization Problem

The following optimization problem is formulated for one NC block i (one-block subproblem). The transition conditions between the blocks are expressed in the boundary conditions for the one-block subproblem (such that no steps in the contour feed between the blocks occur).

Given: A path according to equation 1 and a machine model according to equations 5 and 6.

Solve for: $\dot{s}(s)$, such that

$$I := \int_{t_0}^{t_1} dt \rightarrow \min. \quad (7)$$

The minimization of time t in equation 7 accords to a *maximization of contour feed $\dot{s}$* along the path [6]. Thus the following algorithms simply try to maximize $\dot{s}(s)$ without looking at the cost function 7.

Constraints: The constraint are limited tool feed, actuator feed, actuator force and boundary conditions.

3 Algorithm

To solve the optimization problem over all the NC blocks and the one-block subproblem described in section 2.3, two steps are necessary. They are described in sections 3.1 and 3.2.

3.1 Overall routine

Transitions between path segments defined in the form of equation 1 are at least C^0 -continuous (no steps occur for $\mathbf{x}$). Thus, "corners" and transitions with discontinuous curvature may often appear. To prepare transition conditions for the following one-block subproblem the overall routine has two choices:

- Reducing feed while passing the discontinuity to a low value defined by the programmer,
- substituting parts of adjoining NC blocks by smoothing blocks.

Both strategies are realized in the system described in this paper. Nevertheless, the description of these would go beyond the scope of this paper.

3.2 One-block subproblem algorithm

The algorithm solving the one-block subproblem described in section 2.3 is divided into two parts (according to [6])¹:

1. Finding the maximum allowed *boundary feed*, $\dot{s}_{max}(s_k)$ by setting $\ddot{s} = 0 \forall s_k$,
2. connecting the areas of increasing and decreasing $\dot{s}_{max}(s_k)$ by maximum allowed *boundary acceleration*, $\ddot{s}_{max}(s_k)$.

Hereby, a sampled trajectory $\dot{s}(s_k)$ with sampling index k is regarded.

Step 1: Finding boundary feed. The computation of $\dot{s}_{max}$ is based on the constraints in section 2.3, since all these constraints are functions of $\dot{s}$. Thereby, the actuator force constraint is reduced by setting $\ddot{s} = 0 \forall s_k$. Note that there is no requirement on C^0 -continuity of $\dot{s}_{max}(s)$, since C^0 -continuity will be reached after having done step 2. Especially when actuators have to change their direction of motion, Coulomb friction will change its sign. This may lead to C^0 -discontinuities of $\dot{s}_{max}(s)$.

Step 2: Finding boundary acceleration. After having computed $\dot{s}_{max}$, boundary acceleration $\ddot{s}_{max}(s_k)$ has to be found. Thereto, $\dot{s}_{max}$ is corrected such that the actuator force constraint is achieved. In this step, an increasing $\dot{s}_{max}(s_k)$ is optimized forwardly, decreasing $\dot{s}_{max}(s_k)$ backwardly.

For illustration, figure 4 shows an example. At position (1), the trajectory starts with boundary acceleration, until it reaches boundary feed at (2). The next interval of increasing feed starts at (3). Thus, optimization goes on at this point until it reaches boundary feed at (4). No more interval of increasing feed can be found in this example. Therefore, backward optimization is done by starting at (5), until it reaches the (corrected) boundary feed at (6), etc.

4 Interface to the machine's NC

The computed feed is first divided into adequate parts. These parts are then approximated by a 5th order polynomial with a least-squares procedure using weighting and damping such that:

$$\dot{s}_{opt}(s) = \vartheta_5 s^5 + \vartheta_4 s^4 + \vartheta_3 s^3 + \vartheta_2 s^2 + \vartheta_1 s + \vartheta_0. \quad (8)$$

The coefficients ϑ_i are then stored in a modified NC program together with the given geometric commands.

¹In the following explanations the block index i is omitted.

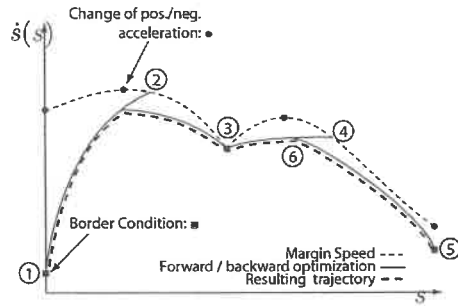


Figure 4: Example for optimization algorithm.

5 Example

The following example has been computed for the *IWF-Hexaglide*. The given tool path is shown in figure 5. The tool first moves in its *distinct* direction

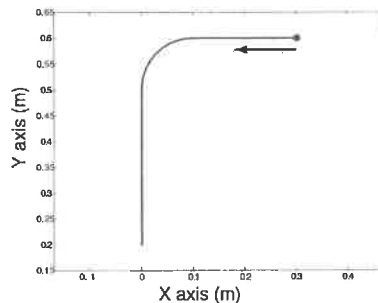


Figure 5: Path example

(-X direction), where the dynamic capacity of the machine does not change². It then moves on a quarter circular arc and moves on in -Y direction.

The resulting feed is shown in figure 6, whereby the dash-dotted curve is the boundary feed computed by step 1 of the one-block subproblem algorithm. Step 2 leads to the solid curve with boundary acceleration, which is the resulting time optimal feed (see section 3.2). In this example no feed reduction over corners or corner rounding is necessary because the tool path is C^1 continuous³.

For comparison the feed for the trajectory in figure 5 has been optimized with both conventional CIC methods (i.e. limitation of speed and acceleration in the tool and actuator coordinate system). The results are shown in figure 7.

²The existence of a distinct direction is typical for PKMs with glide-principle.

³ C^2 discontinuities appear in this path, but they are not yet treated in this work.

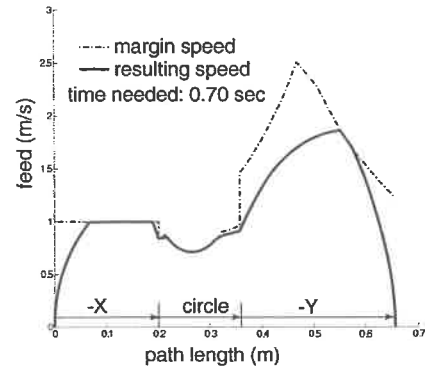


Figure 6: Computed optimal contour feed along the given path.

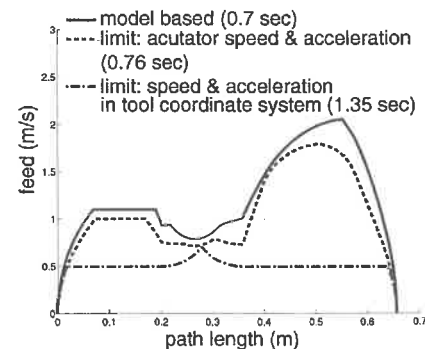


Figure 7: Comparison of 3 methods to get an optimal contour feed.

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FORCE FEEDBACK CONTROL OF TOOL DEFLECTION IN MINIATURE BALL END MILLING

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1 BACKGROUND

Injection molding is an important manufacturing process for optical and mechanical components. The hard steel dies used in this process play a direct role in the quality of molded parts. Traditionally, fabricating these dies involved rough milling followed by heat treatment, grinding, and polishing to the desired shape. Recently, high-speed machining of heat-treated steel (hardness $> 55 R_c$) has become a viable approach for reducing fabrication times while retaining the necessary shape control. However, as feature sizes drop below 1 mm with dimensional tolerances on the order of 10 μm , tool deflection can create significant errors in the shape of mold surfaces. Deflections associated with miniature ball end tools can exceed 30 μm , rendering finished dies out of tolerance.

To date, most of the research associated with milling tool deflection compensation has involved open-loop techniques that use non-dynamic models of the cutting process to modify the desired tool path prior to cutting [2,3,4,5,6]. Research conducted at NCSU's Precision Engineering Center (PEC) resulted in open-loop correction techniques for tool deflection of miniature ball end mills [1]. This effort used a non-dynamic tool force model developed for diamond turning and modified it for ball end milling. These modeled cutting forces were used to predict tool deflections for specified machining conditions, and to modify the tool path before cutting. The specified tool path, together with a CAD model of the workpiece surface, were used to determine the depth of cut, feed rate, and normal cutting force vector. This information was then used to predict the magnitude and direction of cutting forces and tool deflections, which were used to modify the tool path off-line (prior to cutting). Form errors were reduced from 50 μm to less than 10 μm with accurate knowledge of the cutting conditions and parameters.

Despite the promising results obtained from off-line tool deflection prediction and open-loop compensation, these methods rely on accurate models of the cutting process and cannot adapt to changes in model parameters, disturbances in the cutting process, or uncertainties associated with workpiece alignment. Thus, open-loop compensation results are only as accurate as the assumed model parameters and workpiece characteristics. Critical model parameters such as tool wearland, workpiece material properties, and instantaneous spindle speed are difficult to estimate, and may change dramatically during milling.

This paper demonstrates that force feedback can be used to predict tool deflection and compensate for deflection during the milling operation, reducing susceptibility to uncertainties in model parameters and workpiece alignment. A specific force feedback approach is presented: tool deflection prediction (based on a non-dynamic model of tool stiffness). Real-time control algorithms incorporating this method were implemented and evaluated on a high-speed air bearing spindle. A tool stiffness model was developed to predict tool deflections (axial and radial) based on measured forces. Experiments involving machined grooves in hard steel workpieces, including simple slotting cuts and three-dimensional finishing operations, were conducted at various tool tilt angles to evaluate the effectiveness of force feedback control. Results indicate that profile errors can be reduced up to 80% compared to non-compensated cases. These results confirm that real-time force feedback control can significantly improve the dimensional tolerance and accuracy of injection molds created using miniature ball end mills.

2 TOOL FORCES AND DEFLECTIONS

The tool force model developed by Clayton [7] can be used to calculate cutting and thrust forces during milling operations. Inputs to this model include material properties and friction at the rake and flank faces of the tool. Tool geometry and cutting conditions are used to find the cross-sectional area of the chip and the area of contact between the flank face of the tool and the workpiece. These forces rotate with the flank face of the milling tool, but can be readily converted to orthogonal forces (in the x, y, and z machine directions) for comparison to experimental measurements using the three-axis load cell.

The long shank, ball end milling tools used in this research have a 4.0 mm shank with a 0.8 mm ball diameter end. When used to fabricate free-form surfaces, the tool can be loaded in the axial direction, the radial direction, or both. The tool stiffness is significantly lower in the radial direction (see Table 1), therefore regions of a machined surface where the tool is loaded primarily in the radial direction will be subject to large tool deflections and form errors.

Table 1: Measured and computed stiffness values for long shank, ball end tools

Axial Stiffness (N/m)	1421000
Radial Stiffness (N/m)	98930
25 degree Tilt Stiffness - Calculated (N/m)	419565
25 degree Tilt Stiffness - Measured (N/m)	455120

For the experiments presented in this paper, the tool was tilted at a 25 degree angle with respect to the z-axis to emphasize the effects of tool deflection. For arbitrary cutting conditions, the tool stiffness in the direction orthogonal to the workpiece (normal tool stiffness k_n) can be determined by the experimentally validated expression:

$$\frac{F_n}{\delta_n} = k_n = \frac{1}{\left(\frac{\cos^2 \phi}{k_a} + \frac{\sin^2 \phi}{k_r} \right)} \quad \text{where:} \quad \begin{array}{ll} \delta_n = \text{normal tool deflection} & k_r = \text{radial tool stiffness} \\ F_n = \text{normal tool force} & \phi = \text{tool tilt angle} \\ k_a = \text{axial tool stiffness} & \end{array}$$

3 CLOSED-LOOP COMPENSATION OF TOOL DEFLECTION

Real-time force feedback can be used to predict and compensate for tool deflection during milling operations, reducing susceptibility to uncertainties in tool force model parameters and workpiece alignment. Figure 1 illustrates the effect that tool deflection has on a groove profile. The tool tip is programmed to follow a desired path. However, due to deflection the tool tip, the tool actually creates a depth of cut less than desired (labeled "actual depth" in the figure). The shaded area in this figure represents material not removed due to tool deflection.

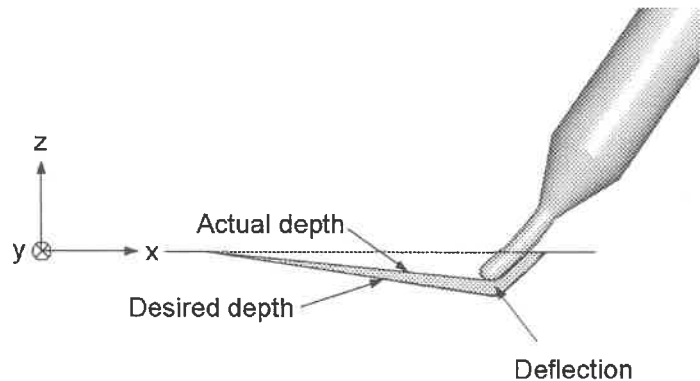


Figure 1: Tool deflection with desired and actual depth

A block diagram of the control algorithm used to predict and correct for tool deflection is shown in Figure 2. The concept behind this algorithm is straightforward: start with a desired tool path, measure real-time cutting force, use the tool stiffness model and measured force to predict tool deflection, calculate an error equal to desired position minus DTM encoder position plus predicted deflection. Once this error has been calculated, PID control either holds position, moves farther into the workpiece in the z-direction increasing the depth of cut, or moves out from the workpiece decreasing the depth of cut. With a validated stiffness model, deflection can accurately be predicted based on tool stiffness and a measured real time cutting force. In this way the system uses real-time force feedback to correct for errors associated with tool deflection and uses encoder feedback to maintain tool path stability. Error calculated by the PID control algorithm is:

$$\begin{aligned} \text{error} &= \text{desired tool position} - \text{encoder axis position} + \text{predicted tool deflection} \\ \varepsilon &= x_d - x + \delta_n \end{aligned} \quad (6)$$

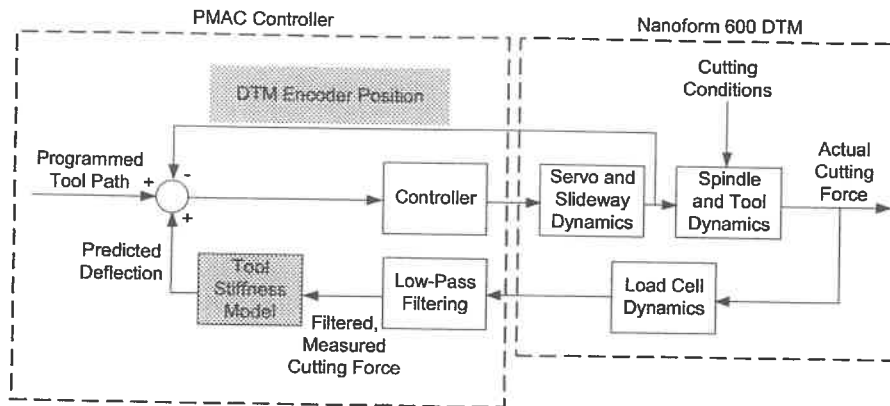


Figure 2: Diagram of predicted deflection algorithm

4 EXPERIMENTAL EVALUATIONS

To evaluate the effectiveness of the force feedback deflection compensation algorithms, extensive machining experiments were conducted on the Nanoform 600 DTM. Groove profiles were machined in S-7 tool steel samples (hardness $> 55 R_c$) mounted to a three-axis load cell. The workpiece was aligned with the axes of the Nanoform and cuts were made along the x-axis (left to right) with the tool tilted at 25 degrees from the z-axis to emphasize the effects of tool deflection (Figure 1).

Slotting cuts with linearly varying depth were made with and without compensation to evaluate the performance of real-time deflection compensation. Slots spanning 20 mm and 0-80 μm in depth were programmed, using a spindle speed of 10,000 rpm and a feed rate of 100 mm/min. This spindle speed was chosen because this was the minimum operational spindle speed to still have enough torque to make cuts in tool steel, and the feedrate was chosen to give a chip load of 5 $\mu\text{m}/\text{flute}$.

Typical results for closed-loop compensation using predicted cutting depth and a non-dynamic cutting force model are shown in Figure 3, with the upper plot representing the uncompensated cutting profile, the middle plot representing the compensated profile, and the lower plot representing the desired profile.

Profile errors were significantly and consistently improved throughout the cutting experiment. Maximum profile errors with predicted deflection compensation are approximately 4 μm , compared to 21 μm with the uncompensated cut. This equates to an 80% reduction in groove profile error. In contrast to the predicted depth

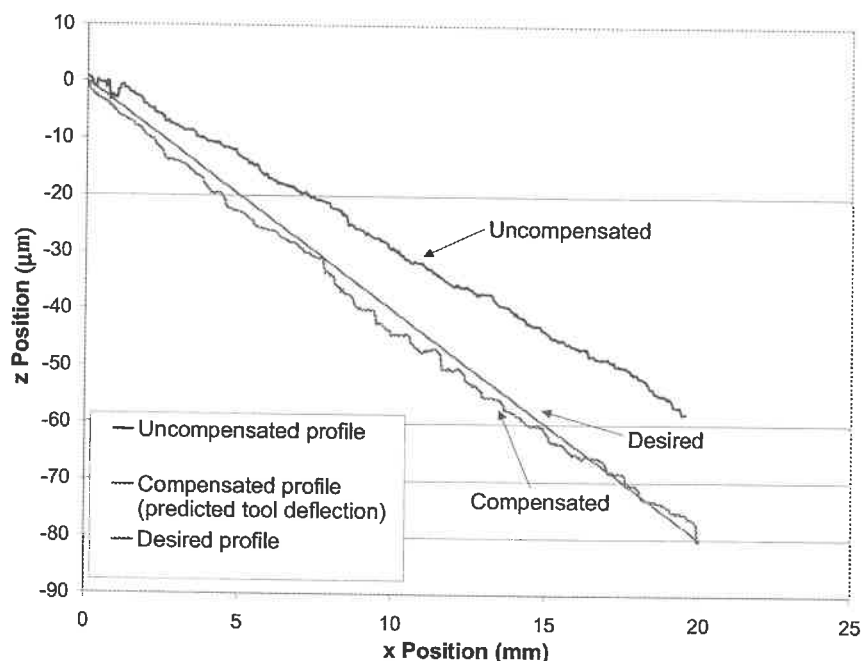


Figure 3: Experimental profile measurements: predicted deflection compensation, linear slotting cut

compensation, errors are initially small and remain small throughout the experiment (between +4 μm and 0 μm), resulting in depth error at the end of the groove of -2 μm .

5 CONCLUSION

The forces generated during milling with a miniature ball end tool are relatively small (less than 10N) because of the limited size and strength of the tool edge. However, tool deflections can be a significant source of profile error because of low radial stiffness. Predicting tool deflection using real-time cutting force measurements can be used to compensate for errors arising from tool deflections and workpiece misalignment. Reduction in error in small groove profiles over non-compensated groove profiles can be as great as 80% depending on the profile.

Overall, force feedback control of miniature ball milling proved applicable in many situations for correcting errors in machined parts due to tool deflection. Significant reductions in profile error for large and small groove experiments were documented, though success depends heavily on the acquisition, processing, and filtering of cutting forces during machining.

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Numerical Control with Surface Driven Interpolation

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Abstract:

A conventional method, used for machining of sculptured surfaces, is that a CAM system generates a large amount of small line segment data and an NC interpolates between the lines. Recently, in order to reduce NC data and to obtain fine surfaces, an NC with the function of NURBS interpolation has been developed. However, since both systems depend on the data that a CAM system generates, the quality of machined surfaces is dependent on the data. Some problems are caused when an NC interpolates the data. This paper discusses such problems from the viewpoint of the relation between a CAM and an NC. Then, an NC with the function of surface driven interpolation is proposed as an ideal system. The conceptual idea, an implementation method and a prototype are introduced. Finally, the merits of the method are discussed and the result of an actual machining is introduced.

Keyword CNC, CAM, interpolation, numerical control, surface machining

1. Introduction

The conventional way of machining sculptured surfaces is shown in Figure 1(a): a CAM system generating a large amount of tool paths and an NC machining of sculptured surfaces using the tool paths. Commercial CAM systems generate tool paths as lines, arcs or spline curves such as NURBS (Non Uniformed Rational B-Spline) curves.

The conventional way has the following drawbacks : A commercial CAM system generates NC data using an individual algorithm respectively for machining sculptured surfaces. Sometimes it generates illegal NC data. For example, a certain CAM system may generate NC data with a slight step even where the surface is smooth. In this case, a numerical control has to perform a reduction process so as to recognize the illegal part and remove it. If a CAM system generates NC data with a slight gap, a numerical control has to fill the gap. A tolerance control may cause NC data without a corner position. Although an NC system can compensate such illegal NC data to some extent, the transaction failure may cause scratches on a surface. It is also difficult for an NC to guarantee an excellent compensation for each CAM system.

A next type of NC system, shown in Figure 1(b), can eliminate such drawbacks that different CAMs cause [1]. Such a type of NC system has an embedded CAM system that can absorb the differences between the CAMs. And, since it does not generate a large amount of NC data, this type of NC system does not need a large storage. In addition, machining conditions can be suitably adjusted for various situations. Since this type of NC generates NC data in real time, an operator can change the machining condition freely when he is machining. It only combines a CAM function with an interpolation function but it is difficult to machine at high speed. The CAM part generates a number

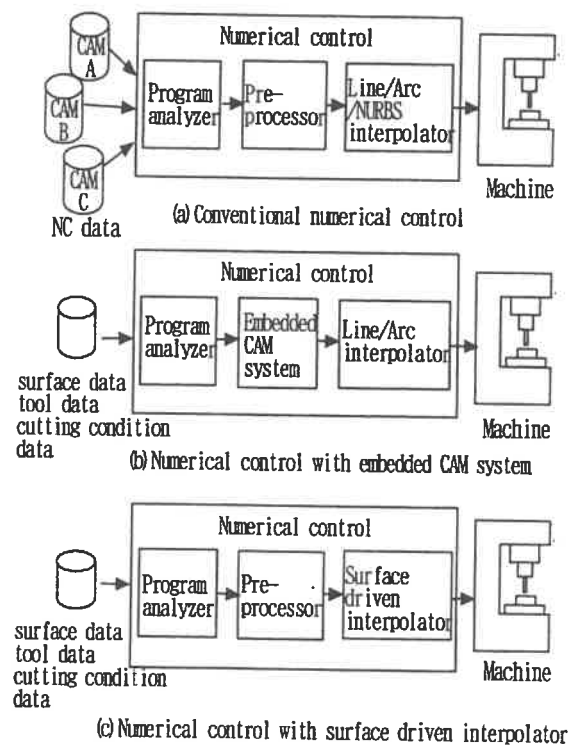


Fig. 1. Three types of numerical control

of small segments to guarantee a given tolerance. The smaller the tolerance becomes, the smaller the segment become and it is difficult for the interpolation part to transact small segments. This causes the speed to slow down.

The reason why such problems are caused is that a numerical control does not have the fundamental function of machining surfaces, but it has the functions of machining of line, arc and NURBS segments. So far, the function of machining of surfaces was obtained by combination with a CAM system.

In this paper, a numerical control with surface driven interpolation is proposed, which interpolates directly from the given surface data without the combination with CAM system. The construction of the numerical control to realize this, the internal algorithm and the machining results are described. The development of numerical control from "point" to "curve" is realized by NURBS interpolation. This paper also shows that the evolution from "curve" to "surface" can be realized by surface driven interpolation [2].

2. Numerical control with surface driven interpolation

2.1 The structure of an NC with surface driven interpolation

Figure 2 shows the structure of a numerical control with surface driven interpolation, which consists of a block of data input, a block of surface evaluation and a block of surface driven interpolation. The block of data input gets surface data, a machining mode, a kind of tool and feed speed. The block of surface evaluation evaluates the condition of a surface and stores the results before interpolation. The interpolation block consists of three parts, namely a surface condition monitor, a speed controller and a position calculator with a collision detector.

The surface-monitoring part generates the speed needed to satisfy a given tolerance and allowable acceleration. The interpolation part calculates a next point Fdt apart from a current position by using a given speed F (dt is sampling time). The collision detector is used to avoid a collision between tool and surface. Interpolation points are transferred into each servo amplifier in real time. Position calculation is important to realize the proposed numerical control. An algorithm to realize that should have the following properties.

1. Fast calculation to guarantee sampling time
2. Robustness and safety for direct machining
3. High precision to guarantee the minimum unit of an NC

A boundary sphere method has been developed [3]. In this method, surfaces are divided into boundary spheres that are represented by a hierarchical tree structure beforehand. First, search leaf spheres that may collide with a tool. Then, mesh points are generated on the part of surface that is included in each sphere and a tool position is calculated by comparing the tool with the mesh points. After applying the algorithm to a CAM system, it was found that this algorithm was fast and precise. Therefore we applied it to the proposed NC.

Simple application of the same module as the CAM system does not give a fine surface. Figure 3 shows tool paths that a general CAM system generates using tolerance control, and ideal tool paths that the proposed NC has to generate. The former paths are generated so that an error may be within a given tolerance "t". However, since the length of the paths is

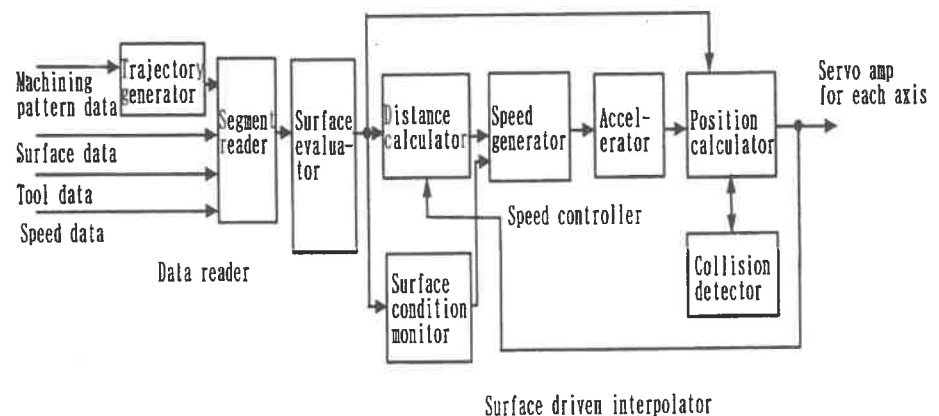


Fig 2 The structure of a numerical control with surface driven interpolation

uneven, it is difficult to drive a machine directly using these paths. The tolerance, which a CAM operator necessarily has to set, is only needed for the algorithm in a CAM system. It is important to recognize that sampling time determines geometric precision. From this, it follows that it is necessary for an NC to generate interpolated points so that a machine can move smoothly as shown in Figure 3(b). The algorithm was investigated to determine whether it is suitable for real time machining and whether the precision is within the sub-micron range.

2.2 Interpolation point generation

It is necessary to control a tool machine smoothly so as not to add any vibration or shock to it. Conventional speed control is classified into two types, namely speed control before interpolation and speed control after interpolation.

Speed control before interpolation controls the set speed, which is an input to the interpolation part. Although this system structure becomes complex, the precision is good. Speed control after interpolation uses a filter for the speed reference after interpolation. Although the system structure is simple, the precision is not good. In this study, we have selected speed control before interpolation in order to achieve high precision. Therefore, we propose a speed control directly and geometrically from surface information without any filter.

Figure 4 shows the method of surface evaluation and speed control. The surface evaluation method looks beforehand at the surface condition along a given trajectory segment. It then calculates the appropriate speed and memorizes it for the given segment. Defining a set speed as F_m , clamp speeds such as from F_a to F_f must be set from a position A to a position F as shown in Figure 4. The part of surface driven interpolation controls feed speed based on the obtained clamp speeds. Since each clamp speed should be strictly kept constant at each position, the feed speed has to be changed beforehand. So, the NC monitors the distance from the current position to the destination and guarantees each clamp speed. Such a process is performed in the distance calculation part and the surface condition part of the monitor.

A feed speed is determined from a set speed, override, emergency stop, etc. Put the current speed of a machine as F_{cur} . After that, the distance L to the next clamp point is calculated. From the distance and the next clamp speed, acceleration mode such as speed-up, speed-down, constant-speed is selected. In the case of speed-down, feed speed is specified according to the distance L . If F is equal to F_{cur} , the speed is kept constant. If $F < F_{cur}$, speed-down is selected by acceleration Acc . If $F > F_{cur}$, speed-up is selected by acceleration Acc . Using the obtained speed, F_{dt} is calculated and the correspondent pulses are transferred to a servo system. When the tool goes beyond the clamp point, a clamp point is renewed. Such a procedure is performed in a sampling time. To calculate a next interpolation point F_{dt} apart from a current interpolation point, an iterative calculation is performed.

3. Machining Result

To prove the proposed numerical control, we used a personal computer (Pentium2 333MHz), a numerical control (Mitsubishi MELDAS M64) and a machining center (MAZAK FJV-20) as the prototype system. The numerical control was modified so that interpolation points generated by the

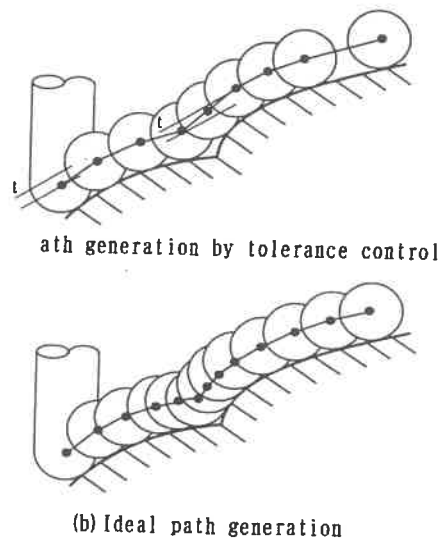


Fig. 3. Path generation

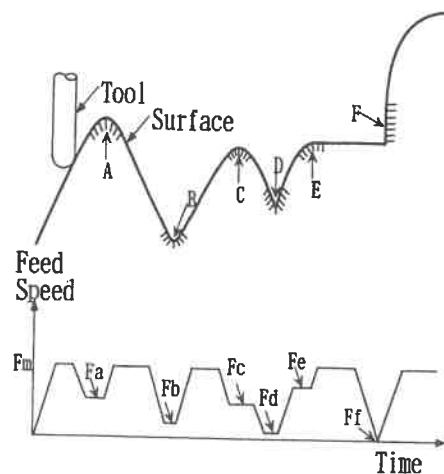


Fig. 4. Surface evaluation and speed control

PC can be transferred to the servo amplifier directly.

Figure 5 shows the results of the interpolation points. We can see a fine speed control at the discontinuous part. In order to machine such a part precisely, the tool speed should be slowed down. The proposed NC can interpolate such a part because the system has surface information and can interpolate in real time. It is difficult for a conventional NC system to do such a control.

Figure 6 shows an example of machining result. Speed control has been performed correctly and fine surface finishes have been obtained.

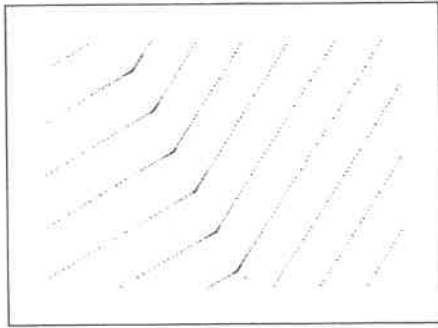


Fig.5 Example of interpolation

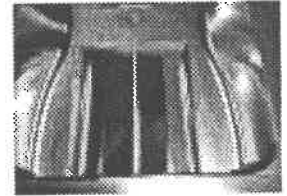
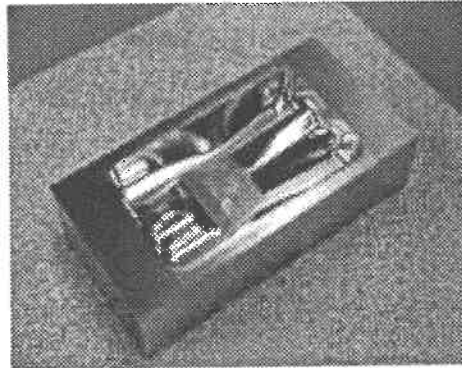


Fig.6 Machining result

4. Conclusion

In this paper, we have investigated and analyzed the problems of conventional numerical control , and have proposed a numerical control method with surface driven interpolation to resolve them. The concepts, the structure of the numerical control and the algorithm have been discussed. We have also evaluated a machining result with a prototype system. From these results the following conclusions were obtained:

- (1) The proposed system gives effective machining by tight coupling of CAM and NC and by discarding an ineffective tolerance control.
- (2) It is not necessary to tune certain parameters of a conventional NC to obtain fine machining, because the proposed method does not need a recognition process inside an NC system.
- (3) High precision machining has been realized even at a discontinuous part, because the system has the information of surface geometry and can generate interpolated positions at high precision even at low speed.
- (4) Since the system does not depend on NC data, which commercial CAM systems generate, the problem caused by illegal NC data is absent.
- (5) The fineness of machining of surfaces has been proved with an actual cutting.

One may conclude that the shift from curve interpolation to surface interpolation is appropriate for the generation of interpolating points that are needed for the machining of any given geometry.

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COMPENSATORY CONTROL OF THERMAL ERRORS FOR HIGH-SPEED MACHINE TOOLS

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Abstract: Objective of this paper is the development of a real-time thermal error compensator for high-speed machine tools. A vertical type high-speed machine tool having a slant column with lightweight moving slides is used to investigate thermal characteristics. The high-speed machine tool has a large thermal drift error within a few minutes. The thermal time constant is only 3~4 minutes at the spindle speed of 25000rpm. Due to tooling problems of the high-speed spindle, the axial offset errors are also appeared in machined specimens. If-Then rules and thermal mode approaches are applied to remove defects of test specimens. The developed real-time thermal error compensator has the compensation period that is adequate for thermal characteristics of high-speed machine tools. The accuracy of machined surface is improved from $70\mu\text{m}$ to less than $10\mu\text{m}$.

Keywords: Accuracy, Axial offset, Compensatory control, High-speed machine tool, Thermal error, Thermal mode, Thermal time constant.

1. Introduction

High-speed machining (HSM) has been attracting interests for many years. Recently, the demand of HSM is increasing in the aircraft industry, automotive industry, die and mold makers, etc. Objectives of the HSM are improvement of surface quality and machining accuracy as well as achieving higher productivity. For these purposes, researches have been continued mainly on the viewpoint of cutting mechanics, cutting tools and machine tools [1-4]. However, there are still many problems to be resolved in the application of the HSM. Current problems include issues of tooling, balancing, thermal behaviors, and reliability of machine tools.

In this paper, thermal characteristics of a high-speed machine tool (HSMT) are investigated through cutting performance tests. The machined specimen has two features that can't be found in conventional machine tools. If-Then rules and thermal mode approaches are applied to remove defects of the test specimen. A real-time thermal error compensator is implemented on an industrial computer. Compensation period of the error compensator is determined to be adequate for thermal characteristics of the HSMT. The accuracy of machined surface is improved from $70\mu\text{m}$ to less than $10\mu\text{m}$ with the compensatory control action.

2. Thermal characteristics of the HSMT

As shown in Fig. 1, the vertical type HSMT having a slant column with lightweight moving slides is used for experiments. Its maximum feed rate is 60 m/min. A spindle with ceramic ball bearings is directly connected to the spindle motor through a coupling. The spindle speeds up to 25,000 rpm.

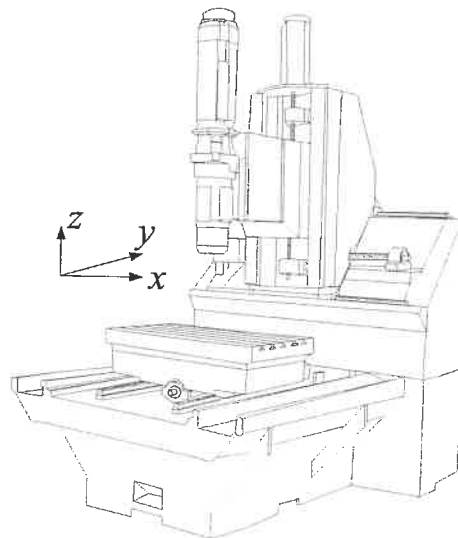
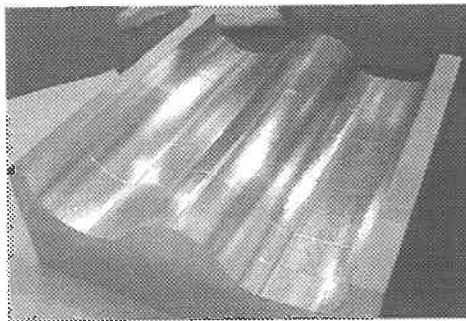


Fig. 1 Vertical high-speed machine tool.

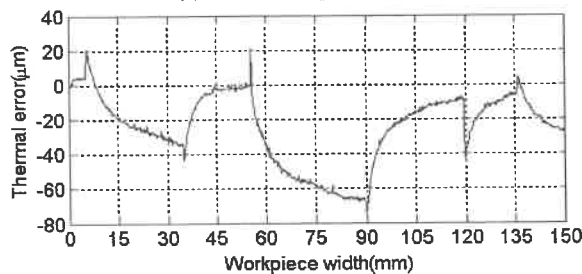
Machining tests with various spindle speeds are performed to investigate thermal characteristics of the HSMT. Table 1 shows the cutting conditions.

Table 1 Cutting conditions.

No.	Spindle speed [rpm]	Cutting width [mm]	Cutting time [min]	Cutting feed [mm/min]
1	5,000	5	11.8	1000
2	20,000	30	30.6	4000
3	10,000	20	29.4	2000
4	25,000	35	32.5	5000
5	15,000	30	35.1	3000
6	5,000	15	35.5	1000
7	20,000	15	15.3	4000
Total		150	190.2	-
Material		Al6061		
Cutting tool		R3.0 ball-endmill 2-teeth, TiAlN coating		
Coolant		Oil Mist		
Cusp		0.42 μm		



(a) Machined specimen



(b) Z-axis

Fig. 2 Result of machining test.

As shown in Fig. 2, the machined specimen has two features that can't be found in conventional machine tools: (1) Discontinuous marks like steps are produced where the spindle speeds are changed. This feature is related with tooling problems of the HSMT adopting the National Taper shank. Centrifugal forces resulted from the high-speeds cause radial expansion of the spindle cone, so that the tool axially displaces itself by the clamping force. The maximum axial offset error is about 40 μm at the spindle speed of 25,000 rpm. There are nonlinear and hysteric relationships between the spindle speed and the axial offset. In order to eliminate this discontinuity of test specimens, If-Then rules are proposed to estimate the axial offsets according to the arbitrary spindle speeds. (2) The thermal growth curve

according to the spindle speed is an exponential form. Due to the high heat dissipation of the high-speed spindle, thermal errors between two discontinuous points are over 80 μm at the spindle speed of 25,000 rpm. This amount of thermal error is beyond the acceptable range of the precision machining. However, the high power cooling units quickly stabilizes the thermal drift errors. It is noted that 95% of thermal drift errors are occurred within 8 minutes at the spindle speed of 25000rpm. Therefore, the HSMT has larger thermal drift errors and much shorter thermal time constants than conventional machine tools. In order to predict thermal drifts of the cutting tool edge, the thermal mode analysis method is applied to the HSMT.

3. Thermal error modeling

3.1 Modeling of axial offset errors

Fig. 3 shows the axial offsets with respect to various spindle speeds.

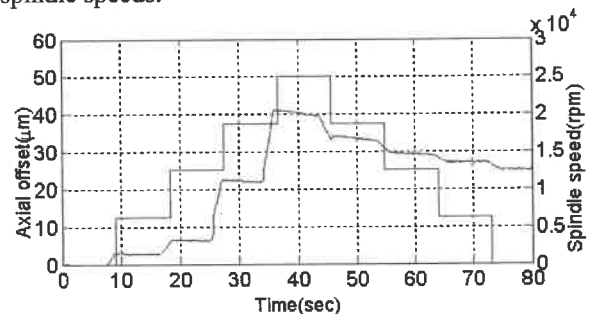


Fig. 3 Example of axial offset errors.

As shown in Fig. 3, there is a residual offset error even if the spindle is stopped. The residual offset is closely related with the maximum spindle speed. The axial offset errors can be modeled with the following If-Then rules.

Rule 0: The maximum spindle speed $r_{\max}$ is continuously updated according to the change of spindle speed.

Rule 1: The axial offset error corresponding to the maximum spindle speed, $\delta_{\max}$, can be expressed as an second order polynomial function.

Rule 2: The residual offset δ_{rsd} can also be expressed as an second order polynomial function.

Rule 3: The axial offset error according to the spindle speed between the maximum and the zero spindle speed, δ_c , can be calculated by using the linear interpolation method between $\delta_{\max}$ and δ_{rsd} .

Fig. 4 shows the flow chart for estimating the axial offset errors based on the above rules.

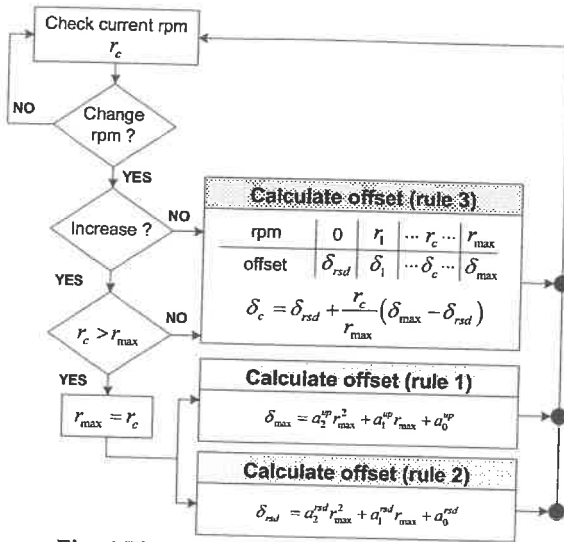


Fig. 4 Flow chart to estimate axial offsets.

3.2 Modeling of thermal drift errors

Thermal mode analysis method [5,6] is applied to the vertical type HSMT. Fig. 5 shows the considered thermal modes for modeling Z-axis thermal drifts.

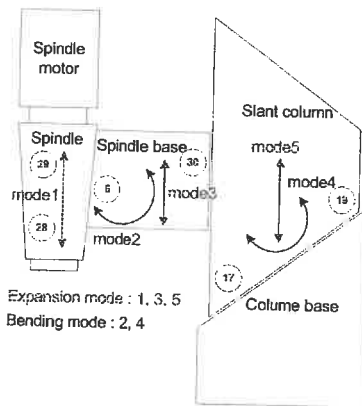


Fig. 5 Thermal modes in Z-axis thermal drifts.

Thermal drifts at the cutting tool edge are modeled as the superposition of thermal expansion and bending modes of each machine component such as a spindle, spindle base and slant column. And, each thermal mode is correlated with temperatures of the corresponding component through a thermal mode gain as follows:

$$\delta_z = G_{\text{exp}}^{\text{spindle}} T_{\text{avg}}^{\text{spindle}} + G_{\text{exp}}^{\text{s.base}} T_{\text{avg}}^{\text{s.base}} + G_{\text{exp}}^{\text{c.base}} T_{\text{avg}}^{\text{c.base}} + G_{\text{diff}}^{\text{spindle}} T_{\text{diff}}^{\text{spindle}} + G_{\text{diff}}^{\text{s.base}} T_{\text{diff}}^{\text{s.base}} + G_{\text{diff}}^{\text{c.base}} T_{\text{diff}}^{\text{c.base}} \quad (1)$$

where, the subscripts *avg* and *diff* mean the average and difference values of two thermocouples, respectively.

In order to determine thermal mode gains, the air cutting tests with various spindle speeds are performed and the non-linear least square fitting method is applied.

4. Implementation of real-time error compensator

The thermal behavior for a specific spindle speed is exponential in nature. As shown in Fig. 6, the compensation process is performed periodically based on a compensation period. If the compensation amount per each compensation period is large, the compensatory control action will leave defects on machined surface such as a scratch or steps.

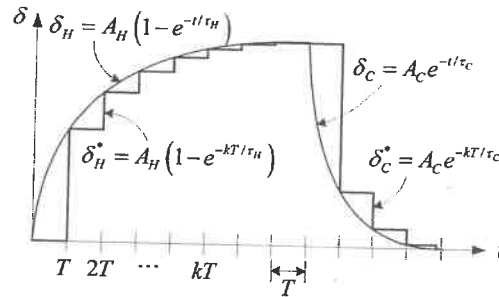


Fig. 6 Periodic compensation processes.

Therefore, in order to prevent deterioration of surface quality due to the compensatory control, the compensation amount per each period must be less than 1 BLU. In this paper, the compensation period of the real-time thermal error compensator is determined from the maximum amount of thermal errors and time constants as follows:

$$T = \min \left\{ -\tau_H \ln \left(1 - \frac{1}{A_H} \right), -\tau_C \ln \left(1 - \frac{1}{A_C} \right) \right\} \quad (2)$$

Since the thermal time constant is small and the maximum amount of thermal error is large, the compensation period for high-speed machine tools is very small relative to that for conventional machine tools.

The estimation model for calculating thermal drift errors is directly related to the sampled temperatures as shown in Eq. (1). Therefore, noises included in temperature data will be amplified by thermal mode gains and significantly deteriorate the surface roughness. The moving average method and digital filter theory are applied to eliminate this phenomenon.

Developed real-time thermal error compensator is realized on an industrial computer. Fig. 7 shows a flow chart for compensatory control of thermal errors. Compensation signals generated on the basis of spindle speeds and temperatures of machine components are transferred to the PLC of the machine tool. The CNC recognizes the compensation signal as the change of the work origin and modifies the current position.

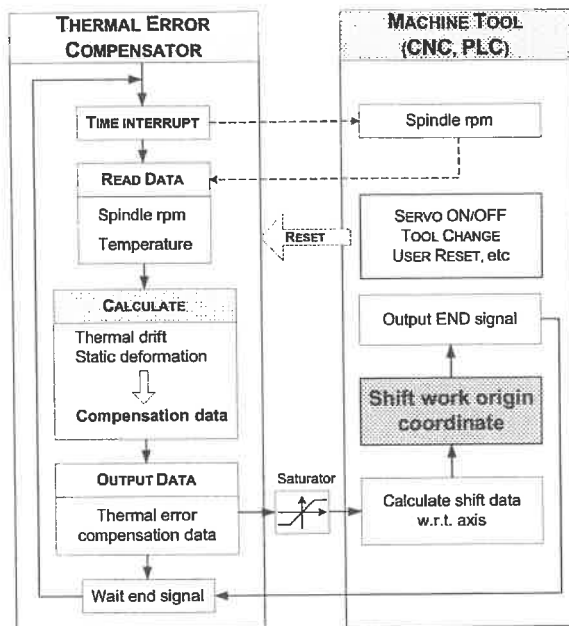


Fig. 7 Flow chart for compensatory control.

5. Cutting performance tests

The efficiency and validity of the developed system are verified through the machining of free-form surfaces with and without compensation. The cutting conditions are equal to the machining test described in Section 2. Fig. 8 shows the results. The accuracy of the specimen is improved from 70 μm to less than 10 μm .

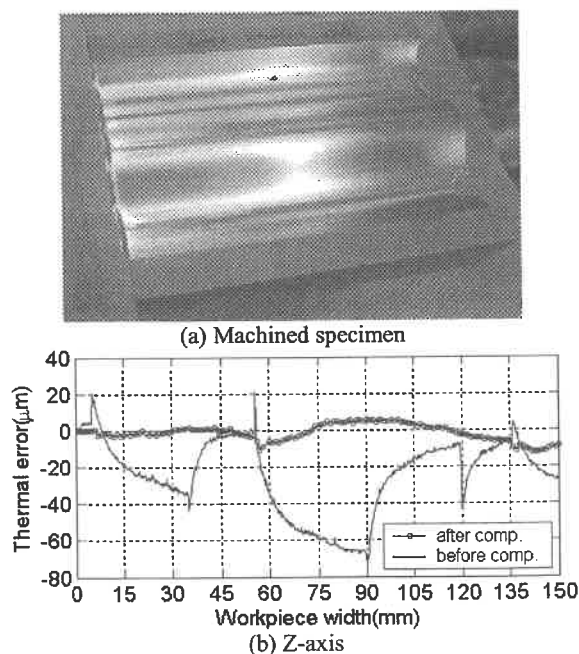


Fig. 8 Compensation result.

6. Conclusions

A vertical type HSMT having a slant column with lightweight moving slides is used to investigate thermal characteristics. Following conclusions are obtained:

1. Due to the high heat dissipation of a spindle and the use of high power cooling units, the HSMT has a large thermal drift error within a few minutes. The thermal time constant is only 3~4 minutes at the maximum spindle speed.

2. The axial offset errors resulted from the tooling problem of the HSMT produce defects like steps on the machined surface. There are nonlinear and hysteretic relationships between the spindle speed and the axial offset.

3. If-Then rules and thermal mode approaches are applied to predict axial offsets and thermal drifts. Thermal drifts at the cutting tool edge are modeled as the superposition of thermal expansion and bending modes of each machine component.

4. The developed real-time thermal error compensator has the compensation period that is adequate for thermal characteristics of the HSMT. The accuracy of machined surface is improved from 70 μm to less than 10 μm with the compensatory control.

Acknowledgement

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Identification of Nonlinear Characteristics for Precision Servomechanisms

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Abstract

A system identification method in the frequency-domain is proposed for a servomechanism with nonlinearities. A linear part of the servomechanism is identified exactly without distortion due to inherent nonlinearities through the proposed method. Parameters of the nonlinearities, including both Coulomb and viscous friction, are identified by the use of limit cycle analysis and describing function approximation. A model-based feedforward controller is applied to verify the availability of the proposed identification method.

Keywords: Coulomb and Viscous Friction, Describing Function, Friction Compensator, Limit Cycle, Nonlinearity, Servomechanism

1. Introduction

The need for high-density magnetic memory devices, processing of semiconductors, and optoelectronic elements has increased the demand for precision servomechanisms. Therefore, identification and compensation of the nonlinearities that seriously deteriorate system accuracy are indispensable to fabricate a high precision servomechanism [1].

In order to identify the nonlinear characteristics of servomechanism, an accurate linear element model should be obtained in advance. Two biased-square wave signals that have different magnitudes and the same frequency are used in identifying the linear element. This identification method does not suffer from the problem of nonlinear distortions and, therefore, is able to provide the accurate model of servomechanism dynamics.

We propose the application of relay feedback experiments to identify parameters of friction model, including both Coulomb and viscous characteristics, of servomechanisms. A relay with time delay is added to the system in order to induce various limit cycle conditions. Then an equivalent nonlinearity can be determined as a combination of the relay and the friction model in the form of describing functions. Using this method, the identification of nonlinearities becomes the identification of describing function parameters using the amplitude and frequency of limit cycle oscillations.

A two-degree-of-freedom controller including a PID feedback controller and a model-based

feedforward friction compensator is applied to verify the effectiveness of the proposed identification method in this paper.

2. Servomechanism with nonlinearity

A. Decouple the servomechanism with nonlinearity

Consider a mechanical part of servomechanism that can be decomposed into a linear element $G_m(s)$ with a nonlinear feedback element $N(v)$ as shown in Fig. 1. The linear element describes the servomechanism dynamics and the nonlinear element describes the inherent friction. In order to identify the servomechanism, an input signal τ_c and an output signal v is used in conventional method with the assumption that the system to be identified is linear [2]. However the actual input signal τ_m to the linear element is quite different from the input signal τ_c owing to effects of nonlinearities.

Consider two square wave signals τ_{m1} and τ_{m2} having different magnitudes and the same frequency as input signals in the identification process. From the linear system characteristics (e.g. communicative law,

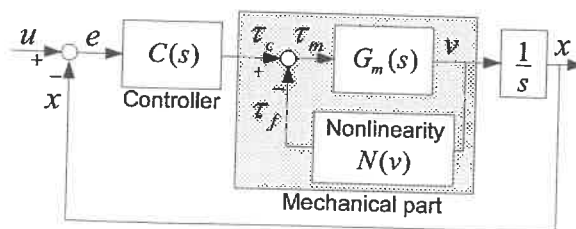


Fig. 1 Block diagram of servomechanism.

distributive law, and so on), the relationship between input and output signals to the linear element is given by

$$G_m(s) = \frac{v_1}{\tau_{m1}} = \frac{v_2}{\tau_{m2}} = \frac{v_1 - v_2}{\tau_{m1} - \tau_{m2}} \quad (1)$$

, where v_1 and v_2 are the output signal corresponding to the input signal τ_{m1} and τ_{m2} , respectively.

In general, the most significant friction components distorting the identification results are the stiction and the Coulomb friction:

$$\tau_m = \tau_c - [F_c \cdot \text{sgn}(v) + F_s \cdot N_s(v)] \quad (2)$$

, where F_c is a Coulomb friction coefficient and $N_s(v)$ is a stiction nonlinearity.

If unidirectional signals which have a proper magnitude are used in the identification process, the stick phenomenon will be eliminated because there are no zero-velocity commands to the system at all times. Then, the difference between two square input signals τ_{m1} and τ_{m2} can be represented as;

$$\begin{aligned} \tau_{m1} - \tau_{m2} &= \tau_{c1} - \tau_{c2} - F_c [\text{sgn}(v_2) - \text{sgn}(v_1)] \\ &= \tau_{c1} - \tau_{c2} \end{aligned} \quad (3)$$

From Eq. (1) and (3), the transfer function of linear element can be described as;

$$G_m(s) = \frac{v_1 - v_2}{\tau_{m1} - \tau_{m2}} = \frac{v_1 - v_2}{\tau_{c1} - \tau_{c2}} \quad (4)$$

Consequently, the linear element shown in Fig. 1 can be exactly identified by taking $(\tau_{c1} - \tau_{c2})$ and $(v_1 - v_2)$ as the I/O pairs, which are equivalent to the I/O pairs $(\tau_{m1} - \tau_{m2})$ and $(v_1 - v_2)$.

B. Experimental Results

We apply the proposed identification method to a precision x-y positioning system. The positioning system is equipped with a ball-screw driving mechanism using AC-servo motors and amplifiers, encoders, digital I/O interfaces, and a PC-based controller.

The input signals τ_{c1} and τ_{c2} composed of Gaussian pseudo-random binary sequence are used as the torque command while motor speed v_1 and v_2 are synchronously recorded. The transfer function of linear element is obtained for the input-output sequence using the MATLAB System Identification Toolbox [3].

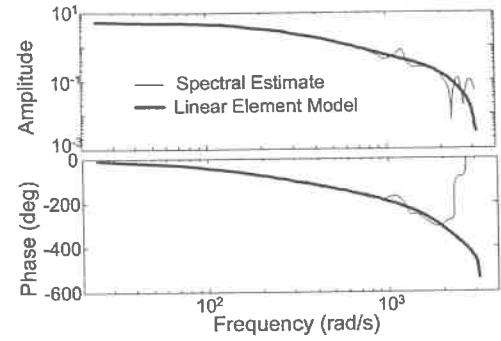


Fig. 2 Frequency response of identified linear element.

For the single axis, a 3rd-order model was obtained with coefficients $f = [1, -1.058, -0.08528, 0.2177]$, $b = [0.05741, 0.2937, 0.181]$. The magnitude and phase response of the identified model are shown in Fig. 2 along with the power spectral estimates. Based on the identified linear element model, parameters of nonlinearities can be accurately determined.

3. Modeling and identification of friction

A. Modeling of friction with describing function

The friction model considered in this paper consists of Coulomb and viscous friction elements as shown in Fig.3. Conventional identification methods require accurate nonlinear models and the information of acceleration or velocity [1,4]. Therefore, relay feedback experiments have been used for identification of friction parameters and frequency characteristics without using the acceleration data [5,6].

A system configuration containing linear element and two relays is devised to identify the friction parameters of the servomechanism. The first relay, RELAY I, is considered as the friction model on the feedback path with the linear element. The second relay, RELAY II, with time delay is added to induce limit cycles as shown in Fig. 3.

The closed-loop configuration shown in Fig. 3 may be modified equivalently as in the configuration of Fig. 4, which consists of the equivalent nonlinearity acting on the linear element. A describing function approximation could be directly applicable toward the

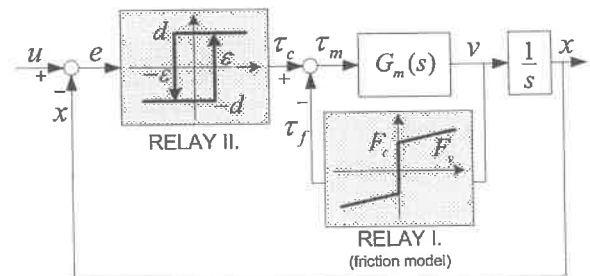


Fig. 3 Block diagram of friction identification.

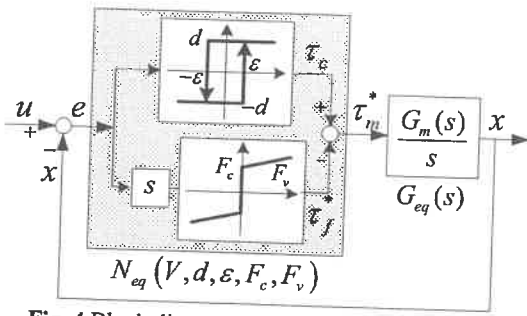


Fig. 4 Block diagram of equivalent nonlinearity.

analysis of the closed loop system. The describing functions of these two relays are given

$$NF = \frac{4F_c}{\pi V} + F_v$$

$$NR = \begin{cases} 0 & , V \leq \varepsilon \\ \frac{4d}{\pi V} \sqrt{1 - \left(\frac{\varepsilon}{V}\right)^2} - j \frac{4d\varepsilon}{\pi V^2} & , V > \varepsilon \end{cases} \quad (5)$$

, where NF is the describing function of the inherent friction model, NR is the describing function of the relay with time delay, V is the magnitude of input signal to nonlinear element, and F_c , F_v is the Coulomb and viscous friction coefficients, respectively.

Then, the describing function of the equivalent nonlinearity $N_{eq}(V, d, \varepsilon, F_c, F_v)$ can be determined as the parallel combination (i.e. the sum) of two describing functions of the two relays with a differential element.

$$N_{eq}(V, d, \varepsilon, F_c, F_v) = NR + jNF$$

$$= \begin{cases} j \left[\frac{4F_c}{\pi V} + F_v \right] & , V \leq \varepsilon \\ \frac{4d}{\pi V} \sqrt{1 - \left(\frac{\varepsilon}{V}\right)^2} - j \left[\frac{4(d\varepsilon - F_c V)}{\pi V^2} - F_v \right] & , V > \varepsilon \end{cases} \quad (6)$$

Using this method, the identification of nonlinearities becomes the identification of describing function parameters using the amplitude and frequency of the induced limit cycle oscillations. Another advantage of this method is that existing control hardware is usually sufficient to identify the friction model and implement its compensation.

B. Limit cycle analysis

The friction parameters can be identified in the equivalent system (see Fig. 4) using the limit cycle analysis [7,8]. The first step of the limit cycle analysis is to find the describing function of the equivalent nonlinearity. The second step is the derivation of limit cycle conditions. The limit cycle conditions can be ex-

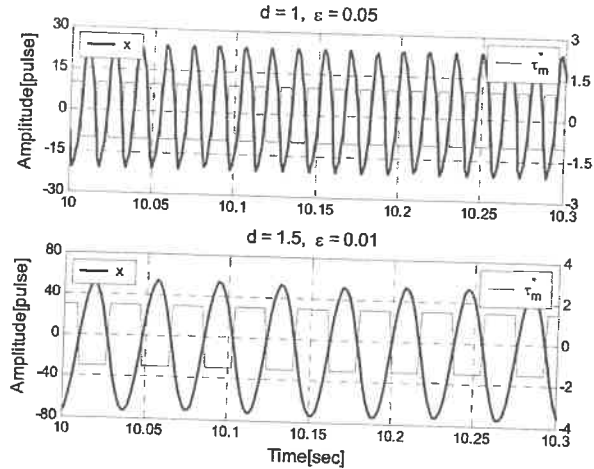


Fig. 5 Limit cycle oscillations by relay experiments.

pressed as follows;

$$1 + N_{eq} \cdot G_{eq}(j\omega) = 0 \quad (7)$$

The amplitude V and oscillating frequency ω of the limit cycle can be approximately given by the intersection of $G_{eq}(j\omega)$ and the negative inverse describing function of the equivalent nonlinearities.

When the describing function $N_{eq}(V, d, \varepsilon, F_c, F_v)$ is unknown and the transfer function of the linear element $G_{eq}(j\omega)$ is known, it is possible to use condition (7) to measure points on the frequency characteristics of the equivalent nonlinearity.

C. Experimental Results

Two relay experiments are conducted in the precision x-y positioning system. By varying delay time and amplitude of the second relay, two limit cycle conditions can be generated. The limit cycle oscillations arising from the two experiments are shown in Fig. 5. Therefore two equations are derived from the Eq. (6), (7) to obtain the unknown parameters F_c and F_v . Clearly, two unknown parameters F_c and F_v can be obtained from the solution of Eq. (8).

$$\alpha_i X - Y = \beta_i, \quad V > \varepsilon, \quad i = 1, 2 \quad (8)$$

$$\text{where, } \begin{cases} X = F_c, \quad Y = F_v, \quad \alpha_i = -4/(\pi V_i) \\ \beta_i = \frac{4d_i}{\pi V_i} \sqrt{1 - (\varepsilon_i/V_i)^2} \cdot \tan[\angle -G_{eq}(j\omega_i)] - \frac{4d_i \varepsilon_i}{\pi V_i^2} \end{cases}$$

The friction parameters identified from two relay experiments are given by

$$\begin{cases} X = f(\alpha_1, \alpha_2, \beta_1, \beta_2) = 0.19801 \\ Y = f(\alpha_1, \alpha_2, \beta_1, \beta_2) = 0.87005 \end{cases} \rightarrow \begin{cases} f_c = 0.19801 \\ f_v = 0.87005 \end{cases} \quad (9)$$

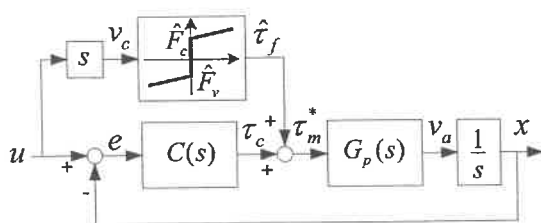


Fig. 6 Feedforward friction compensator.

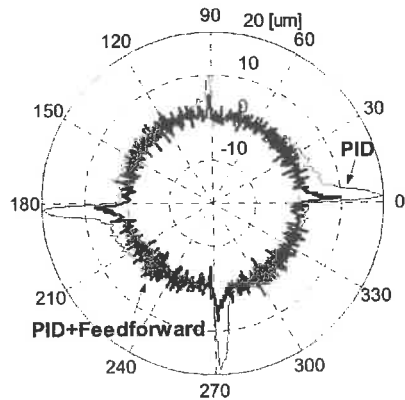


Fig. 7 Position error profile in circular motion.
(Feedrate: 2000mm/min, Radius: 100mm)

4. Friction compensation

A two-degree-of-freedom controller including a PID feedback controller and a model-based feedforward friction compensator is applied to verify the identified friction model. Model-based friction compensators can be classified according to what estimate of velocity is used to evaluate the friction model and what parameters of the entire friction model are applied[1].

In this paper, the command velocity and the identified parameters of Coulomb and viscous friction are used in the friction compensator as shown in Fig 6, where $C(s)$ is the PID feedback controller.

Fig. 7 shows the position error in a circular motion with and without the feedforward friction compensator. Clearly, quadratic protrusion errors and the tracking error in the vicinity of zero-velocity are reduced with the friction compensator. The experimental results indicate that nonlinear characteristics, such as inherent frictions, can be identified accurately by using the propose method.

5. Conclusions

This paper presents a new application of relay feedback experiments in order to identify nonlinear characteristics in servomechanism.

Two biased-square wave signals that have different magnitudes and the same frequency are used in identification process to derive an accurate model of

the system dynamics. This method does not suffer from the problem of nonlinear distortions and, therefore, is able to provide the accurate model of servomechanism dynamics.

A relatively simple identification method of the friction model is presented based on the limit cycle and the describing function analysis.

Experiments have been conducted with identified friction parameters in the precision x-y positioning system. Results from experiments have verified the availability of the proposed identification method

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A Fuzzy Logic Based Adaptive Feedforward PI Controller for Nanometer Positioning

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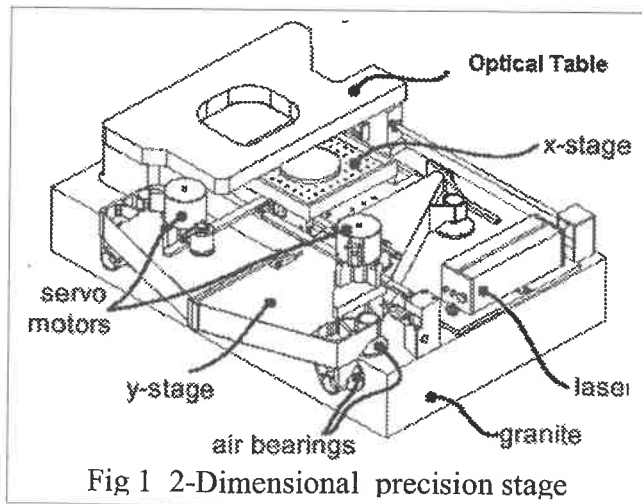
Abstract

A technology driver in high precision manufacturing is a position measurement and control system which allows manufacturers to control the movement of their equipment at extremely high levels of precision. Performances of controllers play important roles in nanometer level positioning. A robust, disturbance resistant controller is necessary for nanometer positioning. However, the time varying and nonlinear properties of plants frequently result in the performance degradation of widely used PID controllers.

The performance of feedback system can be improved considerably by combining PI feedback control with feedforward control together. Feedback controllers are very capable of compensating disturbances because feedback controllers are basically error driven. However, they often suffer from a trade-off between high performance and robust stability. With good dynamic property knowledge available, a feedforward controller may be able to prevent control errors, because feedforward controller output is based on the reference, instead of the error signal. Feedforward controllers improve the control performance by including a feedforward term to the PI output so that the controller can react to the command more quickly in order to bring the plant to the desired setpoint. A fuzzy logic controller will adjust the feedforward and PI feedback gains (K_{ffv} , K_{ffa} , K_{ffd} , K_P and K_I) to adapt the change of dynamic behavior caused by the time varying and nonlinear properties inherent in the positioning systems.

The developed control algorithm is simulated in MATLAB, and is implemented with a TMS320M67 DSP in an experimental nanolithography stage. A X-Y grating based metrology is used to measure the position of stage, which is more robust to environmental change and more suitable with high performance servomechanisms. Experiments show that stage will follow 10 nm step input with nm level steady state error, which is mainly caused by floor and acoustic vibrations.

1. Introduction



A robust and disturbance resistant controller is necessary for nanometer positioning. However, the time varying and nonlinear properties inherent in the system dynamics frequently result in degradation of PID controller performance. For example, air-bearing supported capstan drives are one of several systems used in precision positioning stages (such as the one presented here). They exhibit very low friction and no backlash. However, capstan friction results in non linearity. Open loop testing indicates that the

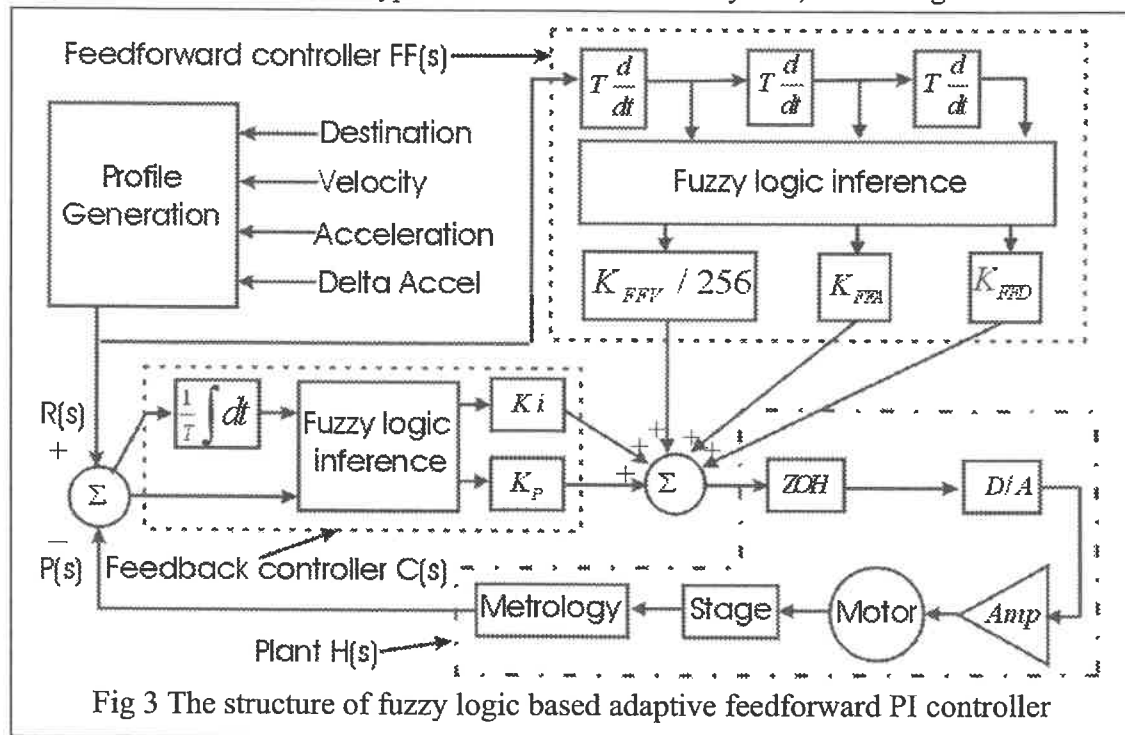
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frequency responses of in macro (1 μm maximum amplitude) and micro (50 nm) range motion are different.

Fixed gain PID controllers are not robust enough and further improvement is needed. Rao and Ro stated that state feedback controllers had better robustness and disturbance rejection capability than PID controllers do. Ro and Hubbel adopted Model reference adaptive control (MRAC) theory in controlling a stage. As a result, separate mathematical models and MRACs had to be developed for micro and macro modes according to the scale of the movement. Although this dual-model MRAC did improve the consistency between micro and macro modes as opposed to a single model MRAC , the problem of deciding the proper threshold for switch from micro to macro mode would make a dual model MRAC controller unsuitable for, say, a 500 nm movement.

2 Design of Fuzzy Logic Based Adaptive Feedforward PI Controller

A block diagram of an analog PI controller with fuzzy logic based adaptive feedforward loop is shown in Fig 2. The PI feedback controller is designed for robustness and accuracy. The integrator increases the type number of the system, allowing it to track



with no (or limited) error. In other words, the integrator will eliminate steady state error to guarantee nm level control accuracy. However, implementing an I-controller by itself will cause instability, and it must be combined with a proportional action to stabilize the system. On the other hand, the PI controller does not help to speed up the system response. Feedforward controllers improve the control performance by including a feedforward term to the PI output so that the controller can react to the command more quickly, in order to bring the plant to the desired setpoint. Note that this function is outside the feedback control loop, and thus does not affect the system's stability. Feedforward helps to minimize following error during motion, by generating most of the DAC output signal from the motion profile rather than the position error.

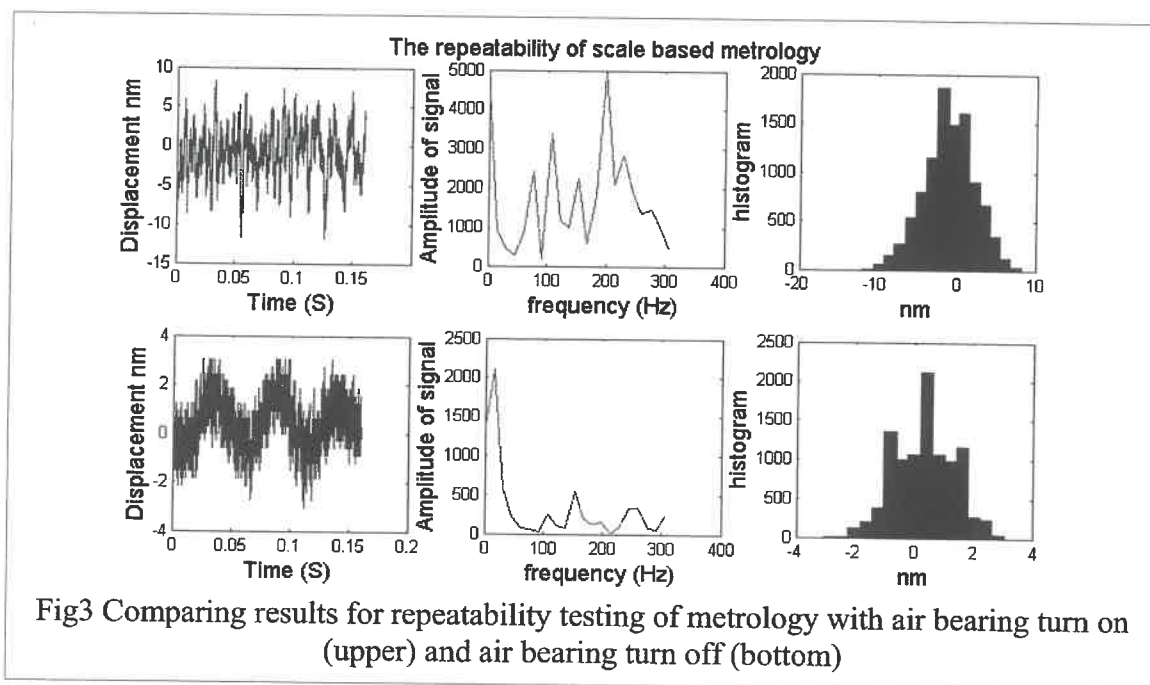
The performance of feedforward and feedback controller depends on the dynamic behavior of plant. Also, feedforward controllers may require large control signals which are beyond the permissible maximum output of the drive. A fuzzy logic controller will adjust the

feedforward and feedback gain (deciding the proper threshold for switching from micro to macro mode) according to the reference gain to deal with those issues.

3 Simulation and Experiment Setup

The proposed control algorithm is simulated in MATLAB. Simulation results show that the fuzzy logic adaptive controller will adapt to change of dynamic behavior caused by the time varying and nonlinear properties inherent in the positioning system dynamics. The testbed for optical Nanolithography which is used to investigate the result of adaptive feedforward controller (shown in Figure 1) is an interferometric lithography stepper, featuring an air-bearing stage, with capstan drive, with HP laser interferometers for feedback. This stage is configured as a stacked-slide X-Y stage. The moving stage is supported by air bearings on the granite surface table.

An XY grid-based planar encoder system is used to measure 2-dimensional ultra-precise



planar displacements of stage, which avoids the turbulence effects which are commonly encountered with laser interferometers or the Abbe errors associated with separate linear scales. The 2-D grid measurement resolution is 0.3nm. The update rate for this metrology is 375 kHz, which is suitable for high performance servo-mechanisms.

The control algorithm is implemented digitally with TMS320C67 digital signal processor which is mounted on an Innovative Integration M67 card. This is a 32-bit standard full-size PCI card. The digital position outputs of the X-Y grating based metrology are accessed through the DSP card parallel digital I/O connector, which is faster than through the PC's bus. The sampling period is 10 μ s. The 16 bit D/A output of the DSP is amplified and then drives the DC motor.

4 Results And Future Work

The measurement noise is shown in Fig 3. When the stage is not floated (resting on granite plate), the amplitude of measurement noise is about 2.5 nm. Comparatively, the amplitude of measurement noise is about 8.5 nm when stage is floated. This may be because the air-bearing is more sensitive to the vibration disturbance from floor or the air pressure in the air bearing changes with time. The controller is tested with different size of step input. The stage will follow 10, 20, 50, 500 and 5000 nm step input with the steady state error around nano-meter level after filtering out the vibration disturbance (Shown in Fig 4).

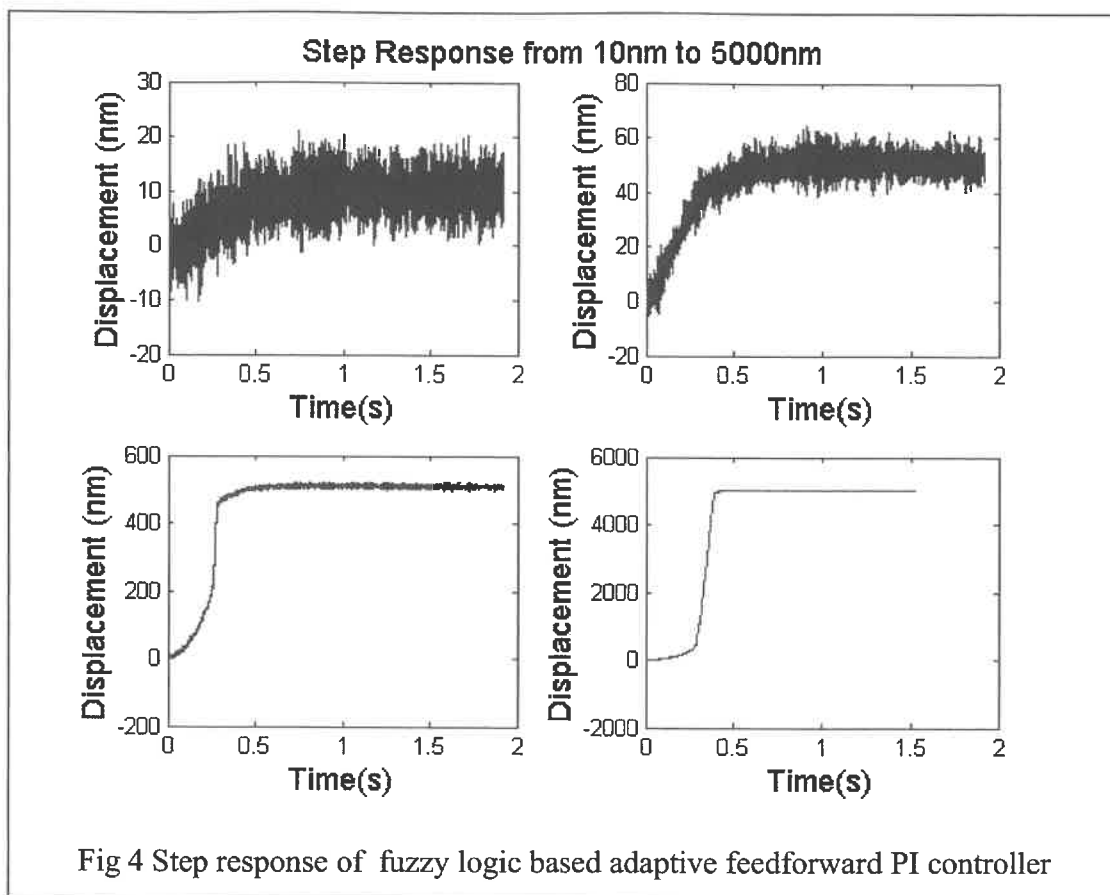


Fig 4 Step response of fuzzy logic based adaptive feedforward PI controller

The PI feedback controller ensures the control accuracy of stage motion. Adaptive feedforward controller improve the transient control performance considerably when there are the changes of dynamic behavior caused by the time varying and nonlinear properties inherent in the positioning systems.

In our current approach, the gain of feedforward controller is determined by trial and error based on experiments. One possible approach in the future is *learning* feed-forward (LFF) controllers. LFF may be able to prevent control errors, because its output is based on the reference, instead of the error signal. They are not necessarily based on a physical process model and are potentially able to learn and reproduce an 'arbitrary' continuous function to any desired degree of accuracy, even if it concerns non-linear and/or time-variant functions.

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Waveform Determination and Positioning Control of the NanoSlider with Inertial Sliding and Stick-positioning

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Introduction

we report on the waveform determination and position-control of a metrological multi-axis nanopositioning device, we called it as NanoSlider, which is operated by piezo-based inertial method, as a sample stage for scanning probe microscopy. We have reported a nanopositioner which has long moving range and high resolution at the same time [1][2][3]. It uses inertial sliding principle and can move in the three-planar direction(XY θ). In the previous report only the experimental movement characteristics and modeling were presented. Dynamic equation formulation was performed and its validity was verified with the experiment. The experimental setup includes the three-axis laser interferometer and mirrors mounted to the body for reflecting laser beam. With the model waveform shape of the input voltage can be determined to improve the performance. The considerations taken for waveform are the oscillation at the end of the actuation step and speed of the movement. The waveform is sectioned by the four segments and each segment is polynomial function or spline curve and the function parameter is varied and the movement performance is estimated numerically with the model. Therefore suitable waveform can be determined from the simulation process. However the nonlinearity and uncertainty exists in the friction phenomena can make the real situation somewhat different from the simulation result, therefore the adaptation in the waveform should be made experimentally when applied to the real system. The modeling of the nanosl原因 includes both the stick phase model and sliding phase model. The stick phase model will be used in the control when stick and theoretical aspects can assist controlling nanosl原因 such as gain tuning and control method (stick-control mode). However it is difficult to use sliding phase model for controlling nanosl原因 in the inertial sliding mode. The reason is that it is hard to implement conventional control theory with that model because the nature of the input is very nonlinear and the motion of the nanosl原因 is quasi-static by nature. Experimental control scheme or many experimental data and its conversion to adequate experimental model will be adopted.

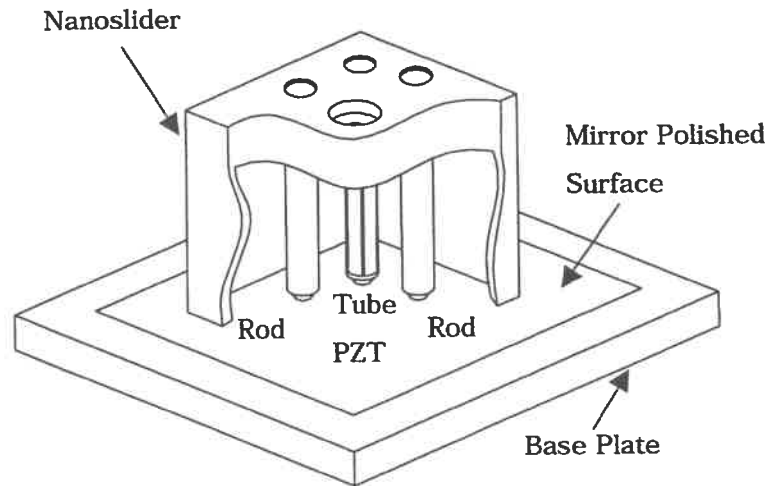


Fig. 1. Nanoslider

Experimental investigation and friction uncertainty region

Many experiments with many input waveforms, such as sawtooth input shows that under certain conditions there are significant friction condition uncertainties. The changes of the position of the nanoslider, moving path history and the motion direction(x and y) make the responses quite different in some cases and the resultant displacement is varied. This uncertain region is characterized by the input voltage and frequency : below certain voltage(about $2\mu\text{m}$) and above certain frequency(200Hz). By the small voltage input, the microscopic structure of the contacting surfaces becomes dominant and by the high frequency input (above natural frequency of the nanoslider in the motion direction) there exist so many sliding-conditions occurring and with the surface character generated by the sliding effect make the responses different (worst case is 100% error). However if the voltage is high enough and the frequency is low enough, the uncertainties become much lower(about 5%) and shows repeatable response characteristics. Therefore the modeling and simulation are based on the responses in this region and positioning control will be operated also.

Waveform determination

With the model waveform shape of the input voltage can be determined to improve the performance. The considerations taken for waveform are the oscillation at the end of the actuation step and speed of the movement. The waveform is sectioned by four segments and each segment is made by polynomial function or spline curve and the function parameter is varied and the movement performance is estimated numerically with the model. Therefore suitable waveform can be determined from the simulation process. However the nonlinearity and uncertainty exists in the friction phenomena can make the real situation somewhat different from the simulation result, therefore the

adaptation in the waveform should be made experimentally when applied to the real system. The simulation uses the modeling equations and the numerical methods are used to gain the responses. Various waveforms varying the curve parameters are put into the real system in order to examine the system responses and input waveform dependence tendencies.

3-axis positioning control

The control methods include two modes of operation which are stick-positioning and inertial sliding motion control. Basically, the inertial control mode uses the experiment-based control scheme. Experiments show that the speed of the nanoslider is nearly proportional to the input voltage and frequency and the speed is quite steady and repeatable. That means when the voltage and frequency are constant the nanoslider moves in the desired direction with almost constant speed. Therefore first the speed control can be used simply. 3-axis planar motion will be controlled simultaneously with the waveform determined and the sawtooth input also. The stick positioning starts when the desired position is near the reach of the nanoslider deflection motion and the final position-control starts.

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Development of Controlled Surface Acoustic Wave Planar Actuators

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Abstract

This paper describes the development of Surface Acoustic Wave (SAW) actuators under closed loop control, which can be applied as ultra precision multi-axis manipulators. In papers presented so far on this topic, linear SAW actuators are driven in open loop. The slider is manipulated by inducing a repetitive small number of wave periods. Although very small steps have been reported for SAW actuators operating in open loop, the reproducibility of such open loop steering may be low in an environment with disturbances. Therefore, we focus on *closed loop control* of the SAW actuator. For this purpose, linear and planar demonstrators have been developed.

If very small steady state speed is required, the wave amplitude becomes smaller than the indentation of the slider and the surface roughness. As a result, a dead band appears in the response of the input signal to the speed of the motor, which makes control difficult. Using this knowledge, an alternative way of actuation has been found on which a patent has been filed: By simultaneously generating waves from opposite sides, the dead band can be eliminated, and the control of a SAW actuator resembles that of a voltage driven DC actuator.

In the absence of waves, the pretension between slider and stator leads to a high contact stiffness. Hence, after switching off power, no additional braking system is required. Once waves are present under the slider, the contact stiffness disappears macroscopically spoken. Instead of stiffness, damping is encountered. Experiments have shown that this applies for both traveling and standing waves. By generating a standing wave in a stator in a particular direction, the slider can be moved across that stator in any direction, without introducing a friction force. Only a damper force has to be overcome. This limits the number of actuators required for a planar actuator.

The planar motor concept proposed features a simple and compact construction with low moving mass, since the drive- and guiding-systems are combined, without the need for external bearings, transmissions or guiding systems. Compared to stacked linear systems, a high stiffness and good dynamic behavior can be achieved. Further experiments are planned to establish the potential benefits of a planar SAW motor in applications of precision engineering, nanotechnology and biotechnology.

Introduction

Conventional multi-axis manipulators are often based on stacked constructions of linear systems comprising an electromechanical actuator, a transmission and a linear guiding system. Such concept results in a relatively large size and a reduced stiffness due to the series connection of components, and hence loss of accuracy and dynamic performance. For this reason, a parallel manipulator, consisting of fewer components would be favorable. In this paper, a direct driven planar (i.e. parallel) manipulator is presented, which is based on SAWs.

SAW actuation principle

The actuation principle of a SAW actuator is based on moving a slider across an elastic solid medium, called stator, through the surface of which Rayleigh waves are propagating [1]. These traveling waves cause elliptical movement of particles at the surface, showing a fixed ratio between tangential and perpendicular motion. The waves are generated by applying an sinusoidal voltage to so-called Inter Digital Transducers (IDTs), finger shaped galvanic patterns applied at a locally polarized stator surface. To generate effective energy transfer, the impedance of the IDT is matched electronically. Our stator is made of PXE43, an isotropic piezo-electric poly-crystal. Considering the speed of Rayleigh waves in our stator and a frequency of 2.3 MHz, the wavelength becomes 0.95 mm. At practical voltages applied to the IDTs, amplitudes in the order of 20 to 40 nm occur, demanding a R_a surface roughness of about 20 nm. By ELID grinding a surface roughness of 10 nm has been attained. In order to generate sufficient traction force to move the slider and to prevent a squeeze air film between slider and stator, the contact surface of the slider (1 cm²) consists of many ball-segments, closely packed, made by lithographic etching of silicon [1]. To achieve high accelerations it is necessary to preload the slider. Depending on the application, several force generating principles can be used, e.g. a vacuum force, a magnetic force, etc.

Linear demonstrator

In our first demonstrator the preload consisted of two sliders moving simultaneously at opposite sides of the stator and preloading each other by a spring, as shown in Figure 1. The required preload force between one of the sliders and the stator is identical to the force between the other slider and the stator. Consequently, the preload force does not load an external guiding system (required to avoid slider motion in y -direction), so the friction is limited. In addition, the preload force does not cause a bending load to the stator in this construction.

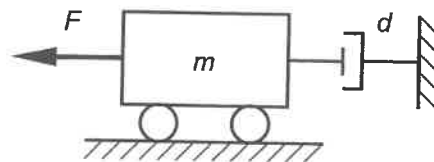
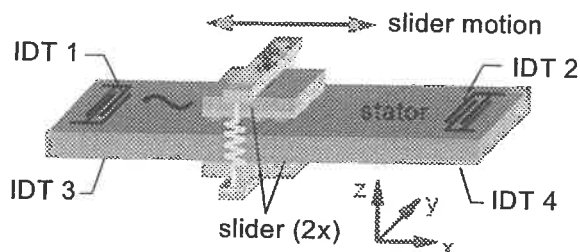


Figure 1: Linear SAW actuator with two sliders, preloading each other.

Figure 2: Model of SAW actuator.

Modeling for control

In papers presented so far, very small steps of 2 nm have been reported for SAW actuators operating in open loop [2]. In an industrial environment, disturbances will harm the reproducibility of this kind of steering. Therefore, we focused on closed loop control of the SAW actuator. Step response measurements showed a first order response from the amplitude of the 2.3 MHz sine to the speed of the slider. This first order behavior can be represented by the simple first order model, given in Figure 2. The damper d is part of the actuator model and the amplitude of the driving sine is proportional to the force F on the mass m .

This system resembles the behavior of a voltage driven DC actuator. For that actuator type (with internal resistance and negligible self-inductance), the damper in the model can be physically explained by a back EMF: for increasing speed, the induced voltage rises. For the SAW actuator, the physical explanation of the damper lies in the 'running-in and -out' of fresh material in the contact between stator and slider. In everyday life, this phenomenon is encountered when driving a car in the presence of cross wind. For a car that is standing still, a constant cross wind leads to a constant displacement of the car due to lateral elastic deformation of the tires (stiffness). But when the car is moving (material is running in and -out of the contact between tire and road), the cross wind leads to constant lateral speed. So although the contact stiffness is still providing the counter force, its macroscopic behavior is that of a damper.

Dead band: cause and solution

The model proposed above suggests a linear system. However, measurements have shown that the linearity between sine amplitude and steady state speed does not exist for very low amplitudes, see Figure 3a. The problem is that if a very small steady state speed is required, the tangential motion must be so small that the perpendicular amplitude becomes smaller than the indentation of the preloaded slider against the stator and the surface roughness of the stator. For amplitudes below this threshold, no slider movement results so this dead band hampers good control.

This problem can be solved if one could use waves for which the relation between tangential and perpendicular motion is not fixed. In that case, the tangential motion can be chosen such that the low speed is achieved, but that the perpendicular motion is large enough. This decoupling between tangential and perpendicular motion can be achieved by using two Rayleigh waves from opposite sides: the longitudinal waves will interfere destructively (if they both have the same amplitude), while the transversal waves will interfere constructively. So the tangential motion is controlled by the difference in amplitude of the two Rayleigh waves, while the perpendicular motion is controlled by the sum of the amplitudes of the two Rayleigh waves. This idea was tested and proved to work, as is shown in Figure 3b. This technique was named 'Dual Side Actuation', or DSA. Since we believe that high performance in closed loop can only be achieved when this dead band is eliminated, a patent application has been filed. When DSA is used, the SAW actuator may now be considered as a linear system, so we can characterize it by a frequency response function (FRF).

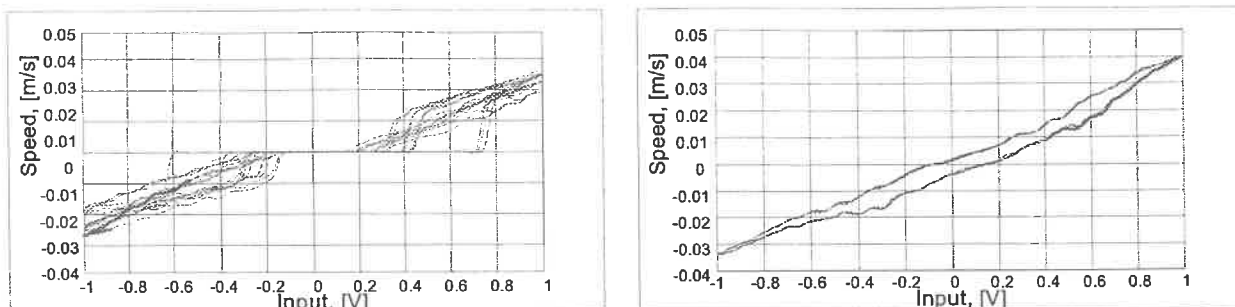


Figure 3: Open loop speed against input voltage, a) dead band present, b) dead band solved.

Note however, the large difference in behavior when the waves are present or when there are no waves. In the absence of waves, the pretension leads to a contact stiffness between slider and stator. Modeling the slider as a mass, this leads to a simple mass-spring system, with low damping. Once the waves are present, the contact stiffness disappears macroscopically spoken. Instead of stiffness, damping is encountered.

It can be concluded that, once the dead band between wave amplitude and response has been eliminated, closed loop control of the SAW actuator resembles controlling a voltage driven DC actuator, which implies applying straightforward control strategies. Because the low order of this mechanical system, a PID controller (maybe with some extra filtering), extended with acceleration and speed feed forward, is a good choice for a controller structure. Using this structure, early test showed a position error in the order of some encoder increments (i.h.c. $0.15 \mu\text{m}$). Principally, the achievable bandwidth of the SAW actuator is limited only by the time it takes the wave to propagate to the slider.

Planar SAW actuator

Using the knowledge obtained from the linear demonstrators, the actuator concept can be extended to a planar SAW actuator [3]. Using DSA and the way of modeling presented earlier, the main issue from a control point-of-view is the coupling between the different degrees of freedom. Our first planar SAW actuator demonstrator (Figure 4, size: $250 \times 250 \times 60 \text{ mm}$) had three stators in which Rayleigh waves can be generated in x- and y-direction. The moving part is a carrier with three sliders, which can be driven individually in x- and y-direction across their accompanying stators. Preload of each slider is performed by (adjustable) magnetic attraction between magnets connected to the carrier and a steel plate under the stators. Position measurement of the carrier is achieved using one planar encoder system, consisting of a grid plate connected to the carrier, and a read head, adjustable with respect to the base frame. Using this encoder reading, planar motion of a carrier is performed by controlling three DOFs: two translations (x and y) and one rotation R_z . In this way the system is over-actuated but not over-determined.

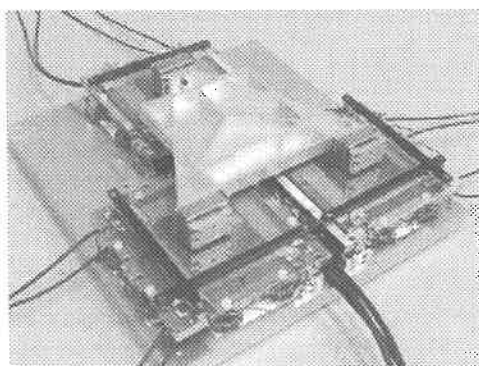


Figure 4: First Planar SAW actuator.

Decoupling is achieved using two transformations: one at the measurement side and one at the actuator side of the controllers. The transformation on the measurement side is such that the position (x, y and R_z) of the centre of gravity of the slider is calculated. Note that this transformation needs absolute position information, so can only be correct after zeroing the incremental measurement system. The transformation on the actuator side (which is distributed over software and electronics in our set-up) is such that all IDTs (12 in total) cooperate to generate force at the centre of gravity in x or y direction and a pure torque. Using this strategy, three SISO (single input, single output) controllers can be used. The decoupling makes it almost like three separate control problems.

The first planar SAW actuator has demonstrated, amongst others, feasibility of generating a planar Rayleigh wave, composed out of two individual perpendicular waves (x- and y-direction) that interfere. This is basically possible since the stator substrate material PXE43 is isotropic. The direction of the composite wave is determined by the ratio of the individual waves, e.g. motion under a 45° angle occurs when both amplitudes for x- and y-direction are equal to each other. In this way, a slider can be driven across the surface in an arbitrary direction. Three of these sliders were used to generate planar motion of a carrier and to constrain in a stiff way the remaining degrees of freedom. In

this over-actuated set-up, *twelve* piezo actuators (IDTs) are required, each matched electronically and each connected to an individual amplifier. One may wonder whether it is possible to drive this carrier with a smaller number of actuators and amplifiers without affecting its behavior, thereby creating a simpler and more cost effective set-up?

Reduction of number of actuators

Considering the set-up as shown in Figure 5, showing the carrier being supported by three sliders A, B and C. Each slider is now driven in one direction instead of two. When the carrier has to be moved e.g. in y-direction, both the sliders B and C are driven. Note that DSA is applied here, so IDTs at both sides of the accompanying stators are energized. In that particular case, moving the carrier implies that slider A has to slide across the stator, requiring a large (static) friction force. Practically, sliders B and C cannot overcome this force and no motion will occur. However, when applying DSA by energizing both IDTs that actuate slider A equally, a standing wave is created for slider A. Whereas no motion occurs in x-direction, the behavior of that slider has changed to that of the mass-damper model of Figure 2, implying that a damper is encountered instead of a friction force (in addition to the dampers felt at the locations of sliders B and C). Generally spoken, by generating a standing wave in a stator in a particular direction, the slider can be moved across that stator in any direction across the stator plane, without introducing a friction force. Only a damper force should be compensated for. Driving the carrier in x-direction is analogous in this set-up, when taking into account the distance from slider A to centre of gravity of the carrier, that has to be compensated by a torque supplied by the sliders B and C. Note that the orientation of the stators can be made such (3 times at 120°) that the driving behavior becomes equal for both x- and y-direction. Experiments have shown that this set-up (Figure 5) is feasible to drive the carrier in three degrees of freedom, requiring only half of the number of actuators and amplifiers compared to Figure 4, without affecting its driving behavior significantly.

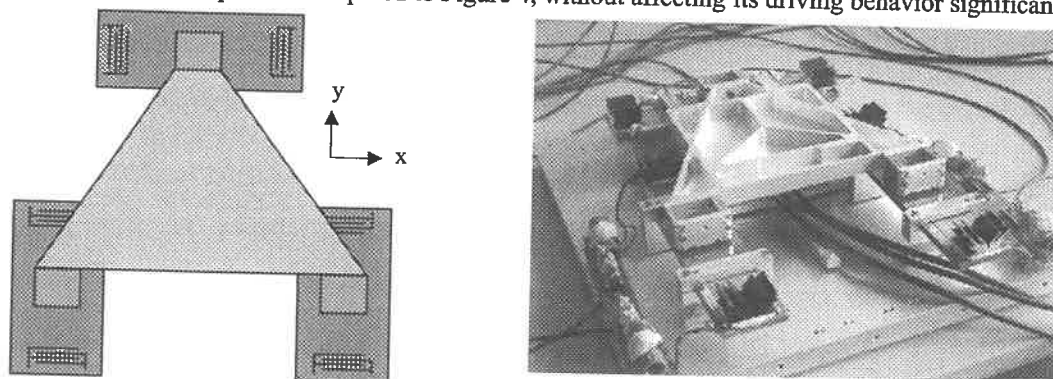


Figure 5: Planar SAW actuator with reduced number of actuators, constructed by three linear SAW actuators.

Conclusions

In this paper the development of SAW planar actuators under closed loop control is described. To investigate the open loop and closed-loop behavior, linear and planar demonstrators have been developed. A different way of actuation has been found and patented, to overcome a dead band in the response of a conventionally driven controlled SAW actuator. In this way, the control of the SAW actuator resembles that of a voltage driven DC actuator. Modeling and experiments have shown that by generating a standing wave in a stator in a particular direction, a slider can be moved across that stator in any direction across the stator plane, without introducing a friction force. Only a damper force has to be overcome. This limits the number of actuators required for a planar actuator thereby creating a simpler and more cost effective set-up. The planar actuator concept proposed features a simple and compact construction with low moving mass, since the drive- and guiding-systems are combined, without the need for external bearings, transmissions or guiding systems. Compared to stacked linear systems, a high stiffness and good dynamic behavior can be achieved. Further experiments are planned to establish the potential benefits of a planar SAW actuator in applications of precision engineering, nanotechnology and biotechnology.

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Differentiating Guidance from Support Functions in Ultra Precision Mechanism Design

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A principal recurring limitation on the positioning precision of a mechanism is friction between sliding or rolling interfaces which both support and guide the motion (ie plain or rolling element bearing surfaces). In many (if not most) instruments and machine tools, its seems to be taken as a design premise that gravitational mass forces on a moving carriage or member shall be supported by static guide-ways.

This putative necessity that support forces should equate to guidance forces is questioned. If such a design constraint is removed, it is shown that, with support forces treated differentially from guidance forces, levels of the latter employed at guide-way interfaces may be considerably lower by design. Examples are given from a range of applications where support force offset techniques have been employed to achieve major reductions in coulomb (interface) friction. These include counterbalance mechanisms, elastic elements, auxiliary 'slave' carriages and drives and partial support using hydrodynamic films, hydrostatic flotation and aeroelastic elements. Examples of each will be given and salient features reviewed. An example will be analysed showing five orders of magnitude difference between the friction associated with the drive force and that required for guidance. In such a regime, reductions in levels of guideway-friction of two orders of magnitude are then quite modest objectives which may be readily achieved.

It is suggested that, using the highlighted 'guidance/support divorce' approach, radical improvement in the design performance of mechanisms and machines is possible for particular applications including wafer-steppers and scanning probe instruments.

For such design improvements in precision positioning systems, it is seen that it is productive to adopt a vicarious approach to a wide range of applications in areas which might not seem to relate to requirements. It is suggested that 'applications-based' categorisation of techniques might constrain such an approach and that perhaps a more lateral approach is needed using a wider intellectual 'trawl'.

If given the opportunity for an oral presentation, questions will be posed as to how these underlying techniques may be more readily accessed by the precision engineering community. In particular, how does one 'teach' graduate students to seek such radical techniques from both within and outside their apparent 'field' of research?

Ultraprecision XY stage with millimetre travel range using leaf-spring mechanism and voice-coil motor

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Introduction

Nowadays, manufacturing technologies such as machine tool and semiconductor manufacturing processes and measuring technologies such as scanning probe microscopes are rapidly processed and stimulate the development of precision positioning devices. Fukada[1] divided current precision positioning mechanisms into two categories, based on different fields of applications. One is the positioning mechanism with a long stroke mechanism, and the other is the fine positioning mechanism. But, precision positioning mechanism with nanometer resolution and millimeter travel range is rarely developed.

This article presents the design and performance evaluation of a compact ultraprecision XY-stage with nanometer accuracy and large travel range. The specifications of XY stage are 2mm travel range and the limited total size of 100mm×100mm×50mm and resolution less than 10nm.

Conceptual design

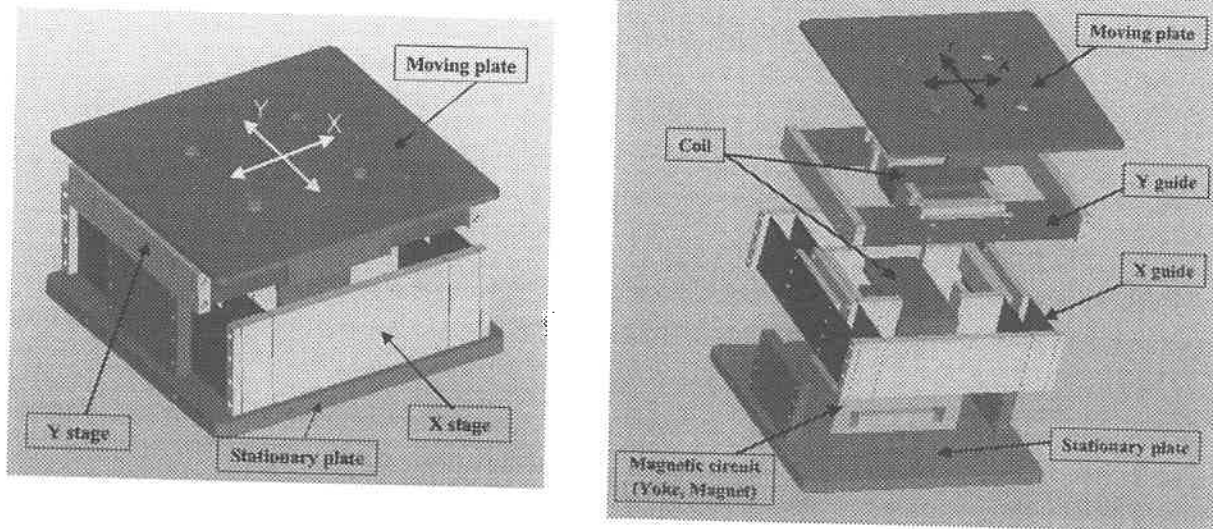


Fig. 1 Structure of proposed XY-stage

The proposed XY-stage is stacked type that Y-stage is piled up on X-stage orthogonally. Two stages are combined with each other by center part wound with coil. Each stage has the

same structure with flexure (leaf-spring) guide whose type is double compound linear spring[2] and VCM (voice-coil motor) actuator. The flexure guide mechanism provides good accuracy due to the characteristics of negligible backlash and stick-slip friction, and smooth, continuous motion. Though monolithic structure is more proper, assembly structure must be used in this system since we can't manufacture a small thickness monolithic leaf-spring guide, which is necessary for large travel range. VCM also has inherently infinite resolution. But the moving part of VCM is different in each stage. X-stage is 'moving coil' type and Y-stage is 'moving magnet' type. Fig. 2 shows the structure of one-axis stage (X-stage).

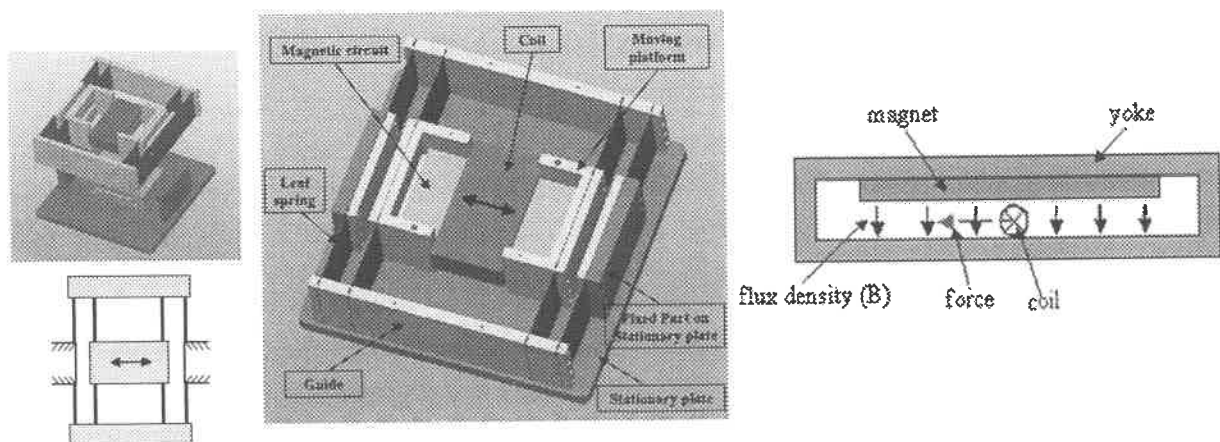


Fig. 2 Structure of X-stage

Modeling & Optimal design

Because the relationship between design variables and system parameters are quite complicated, it is very difficult to set design variables manually. Therefore optimal design is used. The cost function of optimal design is to maximize the 1st natural frequencies of XY-stage. It guarantees fast response.

As optimal design requires static and dynamic characteristics of stage, modeling of stage must be accomplished. Permeance method that gives magnetic flux density at air-gap is used to predict the actuating force of VCM and generalized computer-based method that automatically generates equations of motion and solves them numerically is used to predict static and dynamic characteristics for flexure mechanism. Latter is proposed by Ryu.[3] A little modification of flexure stiffness matrix and pivot point is required to apply his method to leaf-spring mechanism.

For the solution, a sequential quadratic programming (SQP) method and MATLAB have been used. The designed resonant frequencies are 29.169Hz for x-axis and 26.523Hz for y-axis. Fig. 3 shows a designed frequency response function (FRF) of each axis.

Control and Design of a Dual Servo with Fine 6-axis Stage

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Keywords : Dual Servo, 6-axis stage, optimal design, level actuator, VCM, linear motor

Introduction

Many industries require high precision long stroke stages satisfying in-position or scanning stability. For the former, linear motion guide and linear motor are used, but for the latter, air bearing and linear motor do. In addition, dual servo systems are frequently used for ultra-precision stages requiring, high speed, long travel and ultra fine resolution: from hundreds mm to nm. In dual servo system, for in-position stability are used linear motion guide and piezo actuator,[1] and for scanning stability air bearing guide and VCM(Voice Coil Motor) actuator.[1][2] We design a high precision stage for scanning stability with air bearing guide and VCM. Because an optical system requires focusing or leveling system, a bi-directional solenoid actuator is designed and adapted in this paper.[3] In result, this paper describes a dual servo system that is composed of air bearing guide system, VCM, air bearings, and bi-directional solenoid actuator.

Configuration

Fine Actuator, which is a tracker, consists of three VCMs, three preloaded air bearings, and three bi-directional solenoids. And they are symmetrically embodied on a plate as presented in Fig. 1.

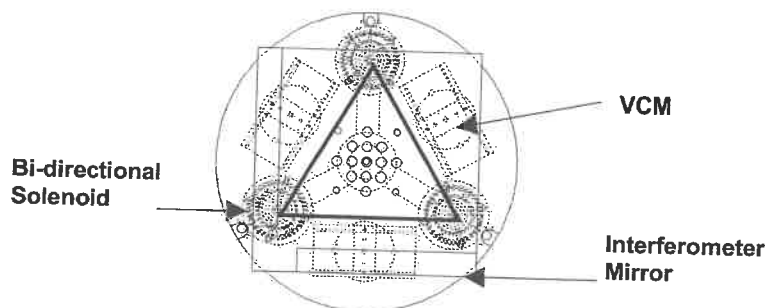


Fig. 1 Triangular symmetric embodied Fine actuator

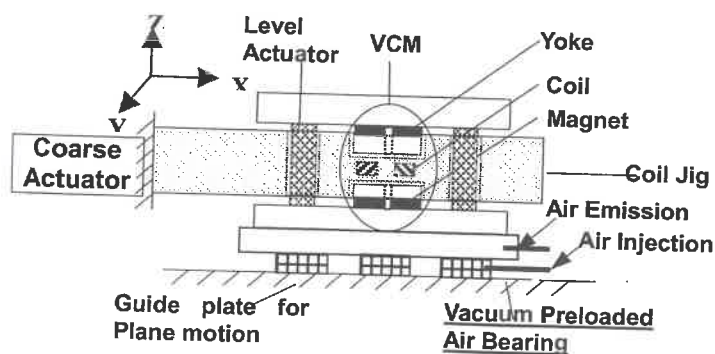


Fig. 2 Sectioned Concept Fine actuator

VCM is a decoupling mechanism in theory, because a coil doesn't have any contact with a magnet and yoke as shown in Fig. 3. Actually, a VCM consists of four magnets, two yoke, and one coil. In this system, force is generated by an energized coil and magnetic field of magnets and yokes.

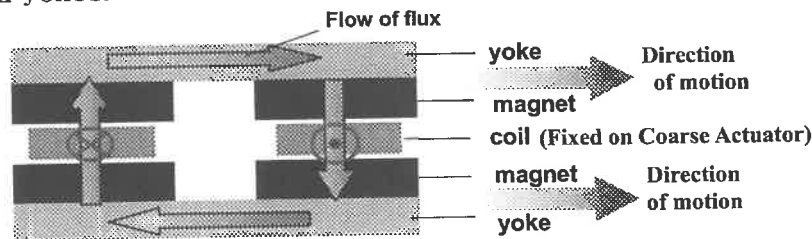


Fig. 3 Suggested VCM structure

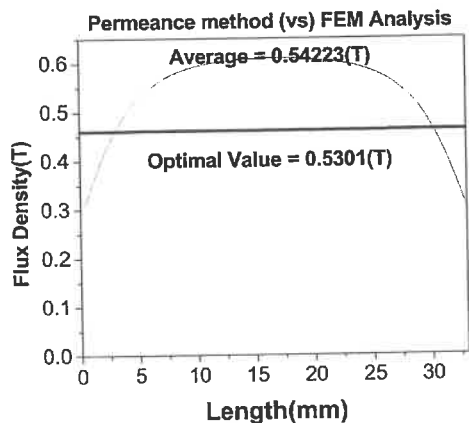


Fig. 5 Comparison between permeance method and FEM

Design Variables	Optimal Values (mm)	Designed Value (mm)
Magnet Thickness	7.5244	7.5
Magnet Length	33.3	33
Yoke Thickness	8.0314	8.0
Coil Thickness	6.8883	6.9
Coil Diameter	0.5922	0.55

Table 1. Design Variables and Optimal Values of VCM

Solenoid for Fine actuator is used for measuring systems and lithography equipments which require leveling motions for DOF(Depth Of Focus) in ZRxRy as shown in Fig. 5. Noble solenoid actuator is composed of two guide springs, one circular magnet, yokes, and solenoid. [3]

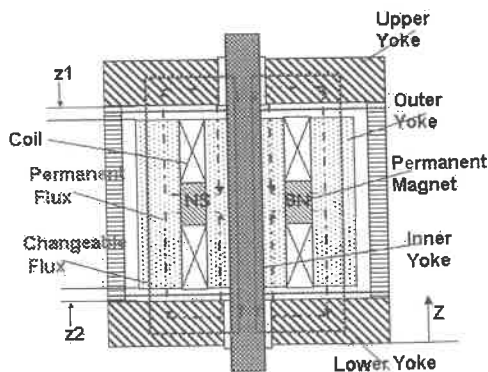


Fig. 5 Suggested Solenoid Actuator

Resistance	Turns	Coil Diameter.	Magnet (mm)	Yoke	Circular Plate(mm)
20Ω	520	$\phi_{coil} : 0.4$ (Bare)	NdFe40 $t : 4$ $\phi_{in} : 15$ $\phi_{out} : 25$	S10C	$t : 0.1$ $\phi_{in_c} : 5$ $\phi_{out_c} : 30$

Table 2. Design Variables of Solenoid

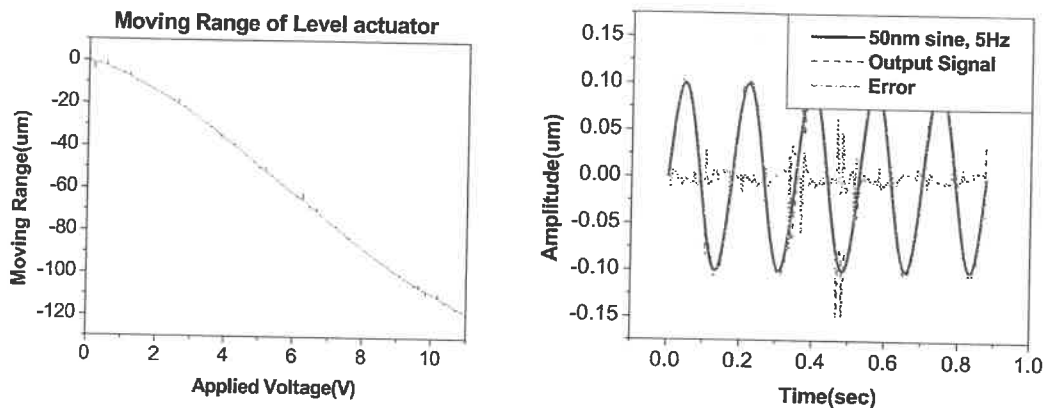


Fig. 6. 100um travel range and 50nm sine wave of a Solenoid

These are embodied in fine actuator plate as shown in Fig. 7.

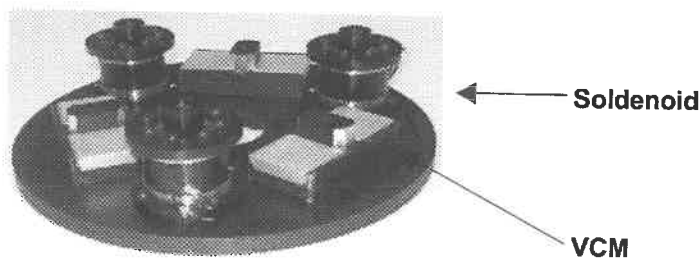


Fig. 7 Manufactured and embodied solenoids and VCMs

Coarse Actuator, which acts a follower, is composed of 4 one sided linear motors using multiple magnet array, a window frame using I-frames, and several air bearings as shown in Fig 8.[4][5]

That is, the follower following the tracker consists of a window frame with four 'I'-beams, a coil jig with three coils to drive Tracker, and several air bearings for its guide mechanism. The advantages of a window frame are symmetric structure, high stiffness, and, to generate high throughput. It is composed of two linear guides with rectangular granites, two magnet arrays, and two coil blocks for four linear motors. The linear motor uses multiple magnet arrays for higher force and less ripple force. The window frame slides along two rectangular granite bars and on a granite plate. At

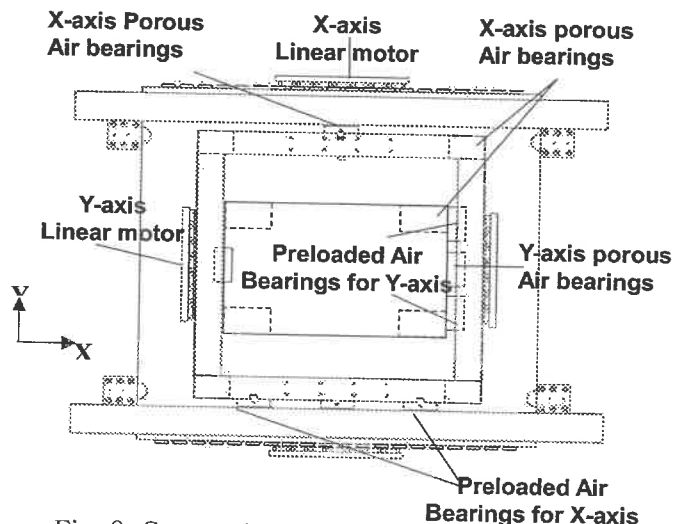


Fig. 8. Coarse Actuator with linear motors, air bearings, and a window frame

that time, it uses several air bearings that some of them are on the side of two rectangular and others on a granite plate. Two of bearings in the rectangular guide are vacuum preloaded types and two not preloaded but ball joint system. This system uses a rectangular granite bar with two bearings as a reference of guide system in one axis that the window frame moves along, and other granite bar with a remaining air bearing as an adjusting guide. And the coil jig, which has three coils for the tracker, moves in window frame and on a granite plate by using forces that are generated by two linear motors. These linear motors are composed of magnet arrays fixed on the window frame and coils fixed on coil jig and operated as the above mentioned. Its guide mechanism with several air bearings is similar to the above window frame's.

At the last, for the long travel about 200mm by 200mm, as the tracker moves, the follower accompanies it by using four linear encoders which measuring motions of linear motors.

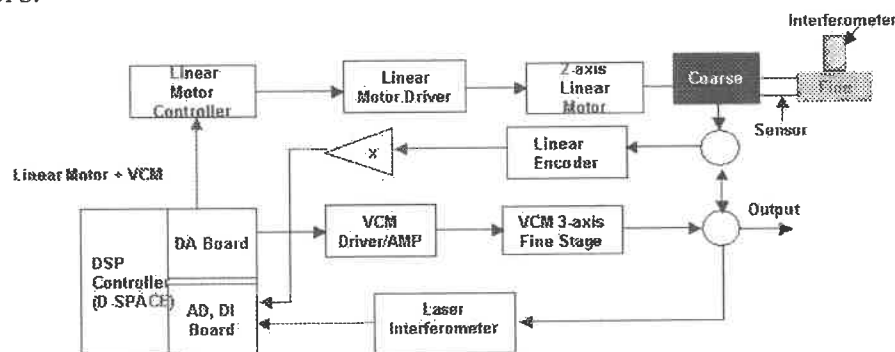


Fig. 9. Control block diagram with Interferometer and Linear encoder

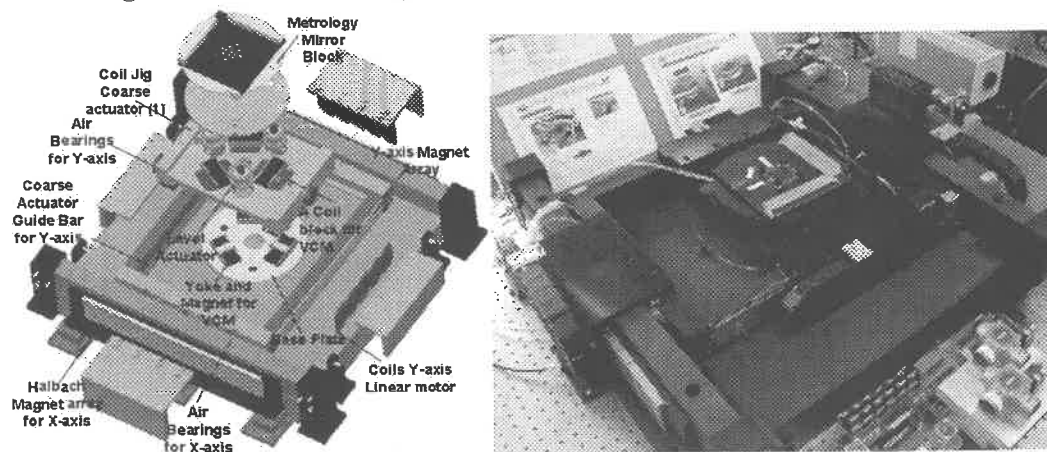


Fig 10. System implementation with a tracker and a follower

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Integrated Design Methodology for High-Precision/Speed Servomechanisms

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Abstract

An integrated design methodology is proposed to ensure the required high-speed and high-precision specifications in servomechanisms, where the interactions between mechanical and electrical subsystems will have to be considered simultaneously. Both the geometric errors referring to Abbe offset and the contour errors referring to circular motions are minimized through the design procedure satisfying allowable structural deformation errors, the relative stability and so on. The validity of the integrated design methodology is confirmed through design results.

Keywords: Abbe error, contouring error, high-precision/speed servomechanism, integrated design, nonlinear constrained optimization, stability

1. Introduction

High-precision/speed servomechanisms have been widely used in the manufacturing and semiconductor industries [1~2]. These specifications significantly exceed the axis motion capabilities of conventional ones. Therefore, it is necessary to devise special design concept to achieve this level of performance for high-precision/speed servomechanism.

In order to ensure the desired specifications in the servomechanism, an integrated design methodology is proposed [3~5], where the interactions between mechanical and electrical subsystems will have to be considered simultaneously during the design process.

In the first step of the integrated design process, it is necessary to focus on the strict modeling and analysis of those subsystems, individually. In addition to the modeling of subsystems, an accurate identification process of the mechanical subsystem is conducted. In the next step, the focus is on integrating subsystems to evaluate the desired performance of the servomechanism. For this purpose, we formulate the optimization problem that includes the relevant parameters of the servomechanism. Finally, simulations and numerical case studies are presented to demonstrate the effectiveness of the proposed design method.

2. Modeling of servomechanism

The mechanical characteristics of servomechanisms such as an equivalent inertia and stiffness have

significant effects on the design optimization.

The free-body diagram of servomechanism to look for dynamic behaviors is shown in Fig. 1. The mathematical model of the mechanical subsystem is constructed by developing the equation of dynamic motion between the motor and mechanical components of servomechanisms. The transfer function between a motor torque τ and rotational motor speed ω is described as Eq. (1).

$$G_p(s) = \frac{\omega(s)}{\tau(s)} = \frac{K_t}{J_m s^3 + a_1 s^2 + a_2 s + a_3} \quad (1)$$

$$a_1 = \left(\frac{K_t K_{emf} + 1}{J_{eq}} \right), a_2 = \left(\frac{J_{eq} K_{eq} \eta + R^2 M_t K_{eq}}{J_{eq} M_t \eta} \right)$$

$$a_3 = \left(\frac{K_t K_{emf} + 1}{J_{eq}} \right) \frac{K_{eq}}{M_t}, b_1 = \left(R^2 + \frac{K_{eq}}{M_t} \right), R = \frac{p}{2\pi}$$

where, M_t is mass of table and workpiece, K_t is a torque constant, K_{emf} is a back-emf constant, η is the efficiency of driving mechanism, p is a ball-screw pitch, and K_{eq} , J_{eq} is an equivalent stiffness and inertia

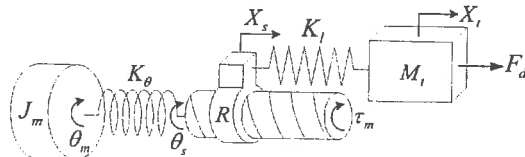


Fig. 1 Free-body diagram of mechanical subsystem.

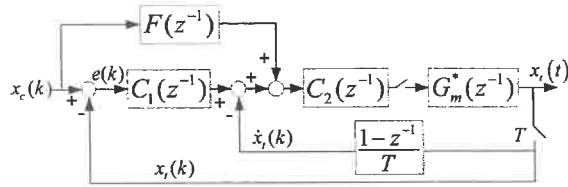


Fig. 2 Block diagram of electrical subsystem.

of mechanical subsystem, respectively.

A two-degree-of-freedom controller that consists of a PID feedback controller and a simple feedforward controller is considered in this paper. The task of the PID feedback controller is to eliminate the steady-state error and to reject the external disturbance such as friction forces. However, feedback controllers have several disadvantages such as poor tracking performance significant overshoots. To overcome these limitations, a direct velocity feedforward controller that adds a velocity command to the velocity feedback loop is considered.

The electrical subsystem structure can be presented as Fig. 2 and Eq. (2), where $F(z)$ is the feedforward controller, $C_1(z)$ is the P-controller as position control, and $C_2(z)$ is the PI-controller as velocity control.

$$F(z) = K_f \left(\frac{z-1}{T_s z} \right), C_1(z) = K_{pp}, C_2(z) = K_{vp} \left(1 + \frac{K_w z}{z-\alpha} \right) \quad (2)$$

From equation (1), (2) and Fig. 2, the open-loop transfer function $G_o(z)$ and closed-loop transfer function $G_c(z)$ of the servomechanism can be represented as

$$G_o(z) = \frac{C_2(z)G_m^*(z)[C_1(z) + F(z)]T_s z}{(1 - F(z) \cdot C_2(z) \cdot G_m^*(z))T_s z + C_2(z)G_m^*(z)(z-1)} \quad (3)$$

$$G_c(z) = \frac{C_2(z)G_m^*(z)[C_1(z) + F(z)]T_s z}{(1 + C_1(z) \cdot C_2(z) \cdot G_m^*(z))T_s z + C_2(z)G_m^*(z)(z-1)}$$

$$\text{where, } G_m^*(z) = (1 - z^{-1})Z \left[\frac{G_m(s)}{s^2} \right]$$

3. Identification of servomechanism

An accurate identification process of the mechanical subsystem is conducted to verify the obtained mathematical model. The model structure for identification is the output error(OE) structure [6] which can be described as;

$$y(k) = \frac{B(z^{-1})}{F(z^{-1})}u(k) + e(k) \quad (4)$$

Parameters of the OE structure given in Eq.(4) are

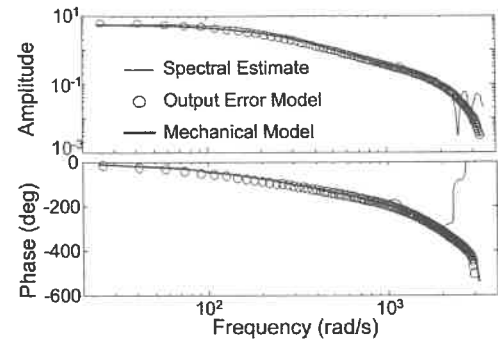


Fig. 3 Frequency response plot of identified system.

estimated using an least squares penalty function into a frequency domain description [6].

$$V(\theta) = \frac{1}{N} \sum_{k=1}^N \beta(k) [\omega(k) - \hat{\omega}(k, \varphi)]^2 \quad (5)$$

In general, an accurate mechanical subsystem model of servomechanisms is difficult to identify due to the nonlinear effects such as Coulomb friction, stiction and backlash. To avoid these effects that corrupt the identification, a biased-square wave input signal is used. The input signal which consists of a Gaussian pseudo-random binary sequence is used as the torque command for the closed current loop while the rotational motor speed is synchronously recorded.

We apply the identification process to a precision x-y positioning system. The positioning system basically consists of a ball-screw driving mechanism with AC-servo motors and amplifiers, an encoder, digital I/O interfaces, and a PC-based controller. A transfer function model is obtained for the input-output sequence using the MATLAB System Identification Toolbox [7].

For the single axis, a 3th-order model was obtained with coefficients $f = [1, -1.058, -0.08528, 0.2177]$, $b = [0.05741, 0.2937, 0.181]$. The magnitude and phase response of the mechanical subsystem model described in Eq.(1) are shown in Fig. 3 along with both the identified OE structure model and the power spectral estimates.

From these identification results, it can be confirmed that the mechanical subsystem model for an integrated design is reliably established.

4. Formulation of integrated design problem

In order to formulate an integrated design problem, the focus is on integrating subsystems to evaluate the desired performance of the servomechanism.

In circular motion, the system input for each axis is given as a sinusoidal wave and the amplitude of the

system output in steady state is the radius of actual circular motion. Therefore the contour error referring to the circular motion is represented as Eq.(6)

$$E_R = 1 - \frac{R_o}{R_i} = 1 - \sqrt{\text{Re}[G_c(e^{j\omega_r T})]^2 + \text{Im}[G_c(e^{j\omega_r T})]^2} \quad (6)$$

Geometric errors should be considered to design high-precision servomechanism. An angular error the worst type of geometric errors in servomechanisms is proportionally amplified by the distance between the central axis of the ball-screw and the table. This results in positioning errors called Abbe errors [8].

In order to guarantee the stability of the designed servomechanism, nominal and relative stability conditions which can be represented in terms of a gain and phase margin must be considered in the integrated design procedure.

A beam model on elastic foundation [9] is considered to obtain the structural deformation errors caused by the workpiece weight and cutting forces. In addition, the critical speed and buckling load of a ball-screw and saturation conditions of electrical sub-

system must be considered in the integrated design procedure. The list of constraints used in the integrated design procedure is shown in Table 1.

Using the modeling and performance analysis described above, we formulate a nonlinear constrained optimization problem including the relevant subsystem parameters of the servomechanism. Design variables $\mathbf{x}$, a multi-object function $F(\mathbf{x})$ and constrained conditions $g(\mathbf{x})$ are defined in this procedure. An Abbe offset, a ball-screw pitch and control parameters of the 2-D.O.F. controller are selected as design variables.

Consequently, the integrated design problem is formulated as the nonlinear constrained optimization problem given in Eq.(7).

Minimize

$$F(\mathbf{x}) = C_1 D_a + C_2 J_{eq} + C_3 E_R + C_4 \frac{1}{\omega_B}$$

subject to

$$g_i(\mathbf{x}) \leq 0, \quad i = 1, \dots, 8$$

$$\mathbf{x}_j^L \leq \mathbf{x}_j \leq \mathbf{x}_j^U, \quad j = 1, \dots, 7$$

$$\mathbf{x} = \{D_a, D_{bs}, p, K_{ff}, K_{pp}, K_{vp}, K_{vi}\}$$

(7)

where, $C_i (i=1 \sim 4)$ is the weighted factor of multi-object function.

5. Results of integrated design

Results of the integrated design (design variables and system performances) are listed in Table 2, 3. And the Bode diagrams and circular motion profiles of the designed servomechanism are shown in Fig. 4 and 5, respectively. The design results not only satisfy all the constraints but also improve the desired system performances.

From the Table 2 and 3, we can see that the Abbe offset and contour error is reduced by 37% and 90% of the initial design, respectively. In addition, it shows an

Table 1 Constraints for integrated design.

Description	Equation
Maximum feedrate	$g_1: V_{\max} - V_{\max}^c < 0$ $V_{\max}^c = \lambda^2 \frac{D_{bs} p}{8\pi L_{sp}^2} \sqrt{\frac{E_{bs}}{\rho_{bs}}}$
Maximum deformation	$g_2: \delta_c - \delta_c^* < 0$ $\delta_c = \frac{\beta}{2k} \left(\frac{2 + \cos \beta a_{ib} + \cosh \beta a_{ib}}{\sin \beta a_{ib} + \sinh \beta a_{ib}} \right) \cdot F_z$
Buckling load	$g_3: F_c^* - P_b < 0$ $P_b = n \frac{\pi^2 E_{bs} I_{bs}}{L_{sp}^2}$
Stability	Nominal stability: $g_4: z_i - 1 < 0$ $z_i = \{z: D_c(z) = 0\}, i = 1 \sim 7$ Gain margin: $g_5: A_m^* - A_m < 0$ $A_m = 1 / G_o(e^{j\omega_r T}) $ Phase margin: $g_6: \phi_m^* - \phi_m < 0$ $\phi_m = \angle [G_o(e^{j\omega_r T})] - \pi$
Saturation	$g_7: \tau_m^{\max} - \tau_m^{\max} < 0$ $g_8: \tau_m^{\max} - T_{\max} < 0$ $\tau_m^{\max} = \frac{J_{eq}}{R} \frac{d^2}{dt^2} a_{\max}$ $\tau_c^{\max} = \max \{ \nu G_{sat}(e^{j\omega_r T}) \cdot X_c(e^{j\omega_r T}) \}$

Table 2 Integrated design results: design variables.

Design variable	Unit	Initial design	Integrated design
$X_1(D_a)$	m	0.0410	0.0257
$X_2(D_{bs})$	m	0.016	0.0125
$X_3(I)$	m	0.005	0.0132
$X_4(K_{ff})$	V/V	0.1	0.1400
$X_5(K_{pp})$	V/V	100	98.6717
$X_6(K_{vp})$	V/V	1	2.9772
$X_7(K_{vi})$	V/V	0.1	0.3974

Table 3 Integrated design results: system performance.

Performance index	Unit	Initial design	Integrated design
E_R	%	6.6800	0.0668
f_B	rad/s	12.717	40.030
J_{eq}	$\text{kg} \cdot \text{m}^2$	1.03×10^{-4}	8.13×10^{-5}
$\tau_{\max}^c$	$\text{N} \cdot \text{m}$	1.29	0.829
A_m	dB	86.828	25.201
ϕ_m	degree	81.749	75.761

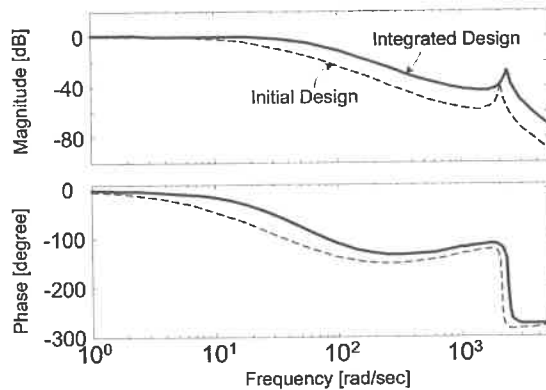


Fig. 4 Bode diagram of integrated designed system.

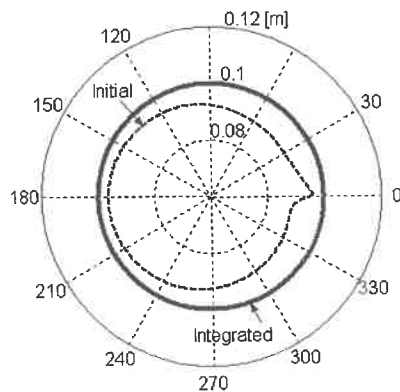


Fig. 5 Circular motion profile for integrated design.

increase of three times over the initial design of a system bandwidth. Therefore a high precision/speed servomechanism has been designed by the proposed design methodology.

The decrease of the relative stability shown in Table 3 means the low-gain of initial electrical subsystems and over design of initial mechanical subsystems.

From these results we can confirm the effectiveness of the integrated design methodology

6. Conclusion

An integrated design methodology is proposed to design a high speed/precision servomechanism. In addition to the strict modeling of subsystems, an accurate identification process of the mechanical subsystem is conducted. Both the geometric error and the contour error are minimized through the integrated design procedure. The validity of the integrated design methodology is confirmed through design results.

Analysis of the simulation results from the integrated design methodology allows better understanding of the dynamic behavior and interactions of the servomechanism. Furthermore, the proposed design methodology offers not only the improved possibility to evaluate and optimize the dynamic motion performance of the servomechanism, but also improves the quality of the design process to achieve the required performance for high-precision/speed servomechanisms.

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CONCEPTUAL DESIGN SUPPORT AND PROTOTYPING OF A MINIATURE MACHINE TOOL

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1. INTRODUCTION

In the former researches [1, 2], the author proposed a new design tool for machine tools combining the form-shaping theory [3, 4] of machine tools with the Taguchi method [5]. The proposed design tool was first applied to miniature machine tools for the "Microfactory" [6] and proved the effectiveness in determining critical design parameters on machine performance. It was also possible to compare several design concepts of miniature machine tools and identify which of the designs had better performance than the other. Since the miniature machine tool designs are not fully supported by existing design experience like conventional machine tools, guidelines for miniature machine tool designs are strongly demanded. Because a machine tool design is a time-consuming process, a design tool that enables the designer to evaluate the machine performance without prototyping must be useful in accelerating the design procedure.

2. CONCEPTUAL DESIGN OF A MINIATURE MILL

2.1 Categorization of design concepts

A machine tool structure can be considered as a chain of rigid components that move against the next components. Tracking the components from the workpiece to the cutting tool, it is possible to categorize machine tool structures by the number of the components that appear before the static part. By expressing transitional motions along the X , Y and Z -axis as 1, 2 and 3, and rotational motions around the X , Y and Z -axis as 4, 5 and 6, major structure types of milling machines that have three transitional motions and one spindle rotation are categorized as Table 1. Among all the structure types, vertical types

were thought to be appropriate for the microfactory, since drilling capability was rather important for the target product. By assuming two horizontal axes have the same characteristics, the candidates were cut down to half. Because two types switching X -axis and Y -axis, such as 12306 and 21306 would have the same performance. In addition, space effective design required that the last transitional motion before the main spindle should be the motion along Z -axis. Determining that "3" comes after 1 and 2, the candidates were reduced to 4 types emphasized in the table.

Table 1 Combinations of axes for a mill

Number of components before the base	Vertical type	Horizontal type
3	12306, 13206, 21306, 23106, 31206, 32106	12305, 13205, 21305, 23105, 31205, 32105
2	12036, 21036, 13026, 31026, 23016, 32016	12035, 21035, 13025, 31025, 23015, 32015
1	10236, 10326, 20136, 20316, 30126, 30216	10235, 10325, 20135, 20315, 30125, 30215
0	01236, 01326, 02136, 02316, 03126, 03216	01235, 01325, 02135, 02315, 03125, 03215

2.2 Design candidates for a miniature mill

As considered in the former section, 4 major structure types were selected as candidates. Fig. 1 shows the outline of the 4 design types. Type 12036 is frequently seen for tabletop drilling machines. Type 20136 is called column-traverse type machine and often used for relatively large product such as automobile parts. A significant question is which of the four typical types has the best

theoretical performance. To calculate machine tool behaviors under various conditions, some control factors representing workpiece dimensions and machine tool sizes were defined. Tables 2 shows the defined control factors. And 15 noise factors to express error sources in machine tools were also defined.

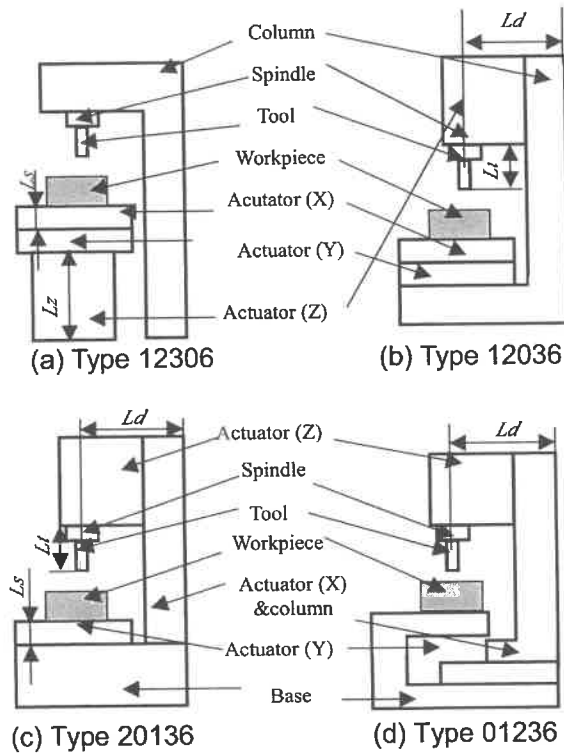


Fig.1 Four major types of mills

Table 2 Control factors and its ranges

Factor name	Variable	Ranges
Workpiece width	w	1, 2, 3, 4mm
Workpiece depth	d	1, 2, 3, 4mm
Workpiece height	h	0.5, 1, 1.5, 2mm
Tool length	Lt	5, 10, 15, 20cm
Spindle - column distance	Ld	5.75, 12.5, 25, 50cm
Height of the vertical actuator	Lz	2.5, 5, 10, 20cm
Thickness of the linear actuators	Ls	1.25, 2.5, 5, 10cm

2.3 Comparison of the machine performance

The form-shaping theory can analyze form-shaping capabilities of machine tools

when the defined control factors vary within the defined ranges and the noise factors affect the performance. By assuming the same control factors and the noise factors to four types, performance of each type can be compared directly. In the calculation, size effects of the noise factors were considered. Figure 2 shows the comparison of the theoretical performances. According to the figure, lines to express the positioning error of type 12036 are always the lowest, and those of type 12306 are the largest. Among the 5 control factors, "spindle-column distance: Ld " is the most critical parameter to affect the machine performance. The figure shows that the design with distributed DOF (type 12036) has better theoretical performance than the design with concentrated DOF (type 12306 or type 01236) has, although the differences between the types are not very large when the machine tool sizes are small.

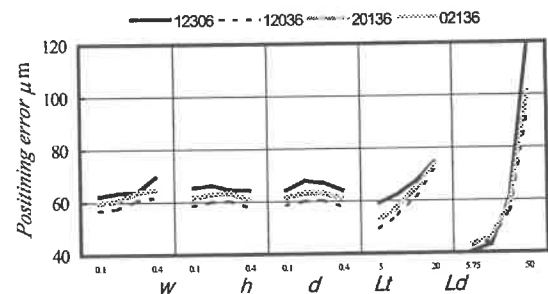


Fig. 2 Comparison of the performance

2.4 Determination of the machine size

The next step was to determine the suitable size for the miniaturization. Basically, positioning error of the machine decreases when the machine tool size represented by " Ld " decreases. However, the optimum size was decided according to the detailed calculation shown in Fig. 3. The vertical axis of the figure shows the mean positioning error and the horizontal axis shows the normalized machine tool size. In this calculation, " Ld ; column-spindle distance" of the normal machine tool was set to 50cm. So, when the index is 1.0, Ld is 50cm. The figure tells that suitable size targeting machining up to 4mm is about 0.075 to 0.125. It means that

column-spindle distance of the suitable miniature mill is about 4 to 6 cm.

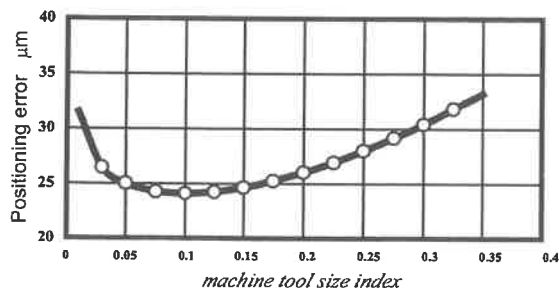


Fig.3 Optimum size for a miniature mill

3. PROTOTYPING OF A MINIATURE MILL

3.1 Comparison of two designs

As it was calculated in the former section, type 12036 was predicted to have the best performance among the 4 major types, and the suitable size represented by "column-spindle distance" is 4 to 6 cm. Actual miniature mill was designed, based on these results. To confirm that the predicted results are reliable, a contrast model for measurement was also prototyped. Fig. 6 is the sketch of the selected design, and Fig. 7 shows that of the other model which was predicted to have worse performance. Both machines have 57.5mm column-spindle distance, according to the optimum size decided by Fig.3.

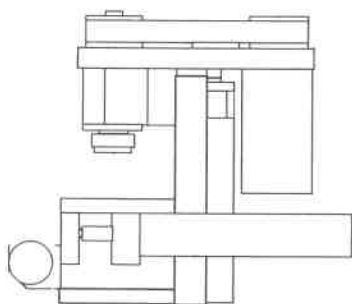


Fig.4 Miniature mill design (type 12036)

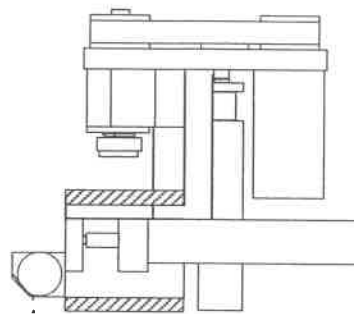


Fig.5 Miniature mill design (type 01236)

Fig. 6 shows the actual miniature mill designed in Fig. 4. The machine size being approximately 12x12x10cm, it can perform drilling up to 2mm in depth, and surface milling up to 4x4mm in area. For the transitional motions and the rotational motion, DC servo motors were used.

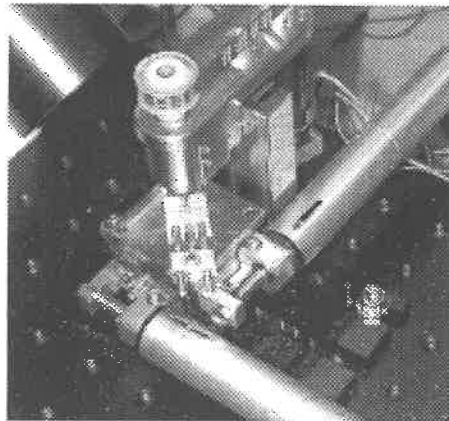


Fig. 6 Prototyped miniature mill (type 12036)

3.2 Measurement of actual errors

The figure showed in the former section tells that the distributed DOF type has a higher performance than the concentrated DOF type. However, to prove the effectiveness of the proposed design tool, it is necessary to compare the measured value and the calculated value. Fig.7 and 8 shows the result of the comparison of calculated and measured positioning errors of the two machines. Straight lines indicate calculated errors. Horizontal lines show Y value of the error vectors, while the vertical lines show X value. Calculated values of the figures indicate that the area surrounded by the vertical line, the horizontal line, X -axis and Y -axis is the

predicted area where the measurement data expected to exist.

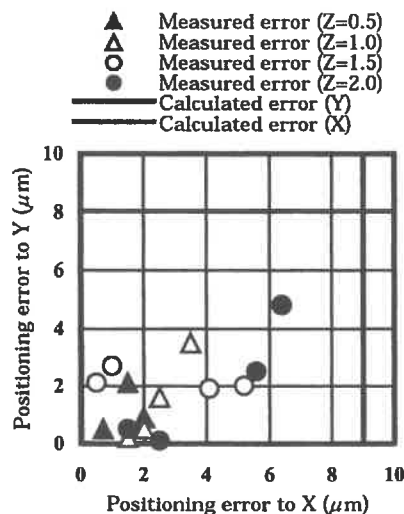


Fig.7 Comparison of measured and calculated errors for type 12036 machine

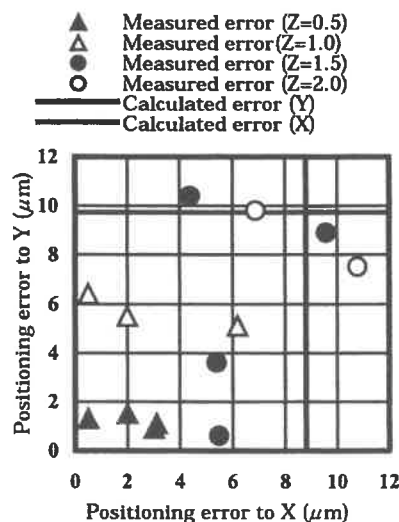


Fig.8 Comparison of measured and calculated errors for type 02136 machine

4. DISCUSSION

The results shown in Fig.7 and 8 indicate some measurement data for the type 01236 exceeded the predicted limit. However, most of the measured data exist within the predicted range. Although the accuracy of the calculation is not enough for predicting machining tolerances accurately, it is sufficient for usage in the conceptual design

stages. Because the purpose of conceptual design stages of machine tools is to decide basic design concept and determine whether the concept will be sufficient for the objective machining. Comparison of Fig. 7 and 8 basically indicates that the type 12036 has better performance than the other. These results enable us to conclude that the proposed design evaluation tool will be useful to assist conceptual designs of miniature machine tools.

5. CONCLUSIONS

- Proposed design evaluation tool was effective to identify the proper design from the design candidates of miniature mills.
- Predicted values of positioning errors were accurate enough for usage in conceptual design stages and the design evaluation results were confirmed by prototyping.
- Among major structure types of miniature mills, distributed DOF type that has two motion axes before the machine tool base, has the best theoretical performance.
- To manufacture small mechanical products up to several mm, about 1/10 miniaturization is likely to be appropriate.

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WATER ENERGY DRIVE SPINDLE SUPPORTED BY WATER HYDROSTATIC BEARING FOR ULTRA-PRECISION MACHINE TOOL

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Key words: Advanced Spindle, Hydraulic Built-in Motor, Water Hydrostatic Bearing, Ultra-precision Machine Tool

1. Introduction

Ultra-precision machine tools are essential for machining precision parts such as molds for various plastic lenses, polygon mirrors for laser printers and so on. Precise motions of the machine tools; i.e., rectilinear and rotational motion; are very important requirements to attain precision machining, in addition to an advanced control algorithm for machine tool control. For the advanced control issue, Nakao and Dornfeld have presented an effective control algorithm^[1]; which is termed the position and AE dual feedback control; for diamond turning machine. It has been shown that the control technique is effective to improve an accuracy of the geometric generating motion of precision machine tool.

This paper presents a new spindle to attain precise rotational motion. The present spindle effectively uses water flow. That is, the water flow can be used for rotating the spindle; i.e., built-in motor; supporting the spindle; i.e., hydrostatic bearing; and cooling the spindle. This paper first describes principles and structure of the spindle. Then performances of the designed spindle are discussed through theoretical and experimental studies.

2. Principle and structure of proposed spindle

Figure 1 illustrates the spindle structure proposed in this study. This spindle has three features, which all use water flow effectively. The features are described in this section.

2.1 Motor function (Hydraulic built-in motor)

First feature of the spindle is that rotational motion of the spindle is generated by water flow energy^{[2], [3]}. In order to rotate the spindle, simple flow channels are designed in the spindle body so that water flow in the channel can generate torque for rotational motion. Pressurized water is supplied from the outside of the spindle into a flow channel; named the main channel; which passes through the center of the spindle in axial direction. After water flows the main channel, the water is discharged from the main channel to outside of the spindle, again, in order to generate torque for spindle rotation. For generating the torque, bend-formed channels, termed the flow-out channel, are designed at two cross sections of the spindle. As illustrated in Fig. 1, the flow-out channels are located between the main channel and the outer surface of the spindle. Flowing water through the flow-out channels, angular momentum of the water flow alters significantly. Therefore, torque to rotate the spindle can be generated, which is based on the angular momentum theory. It is noted that the simple flow

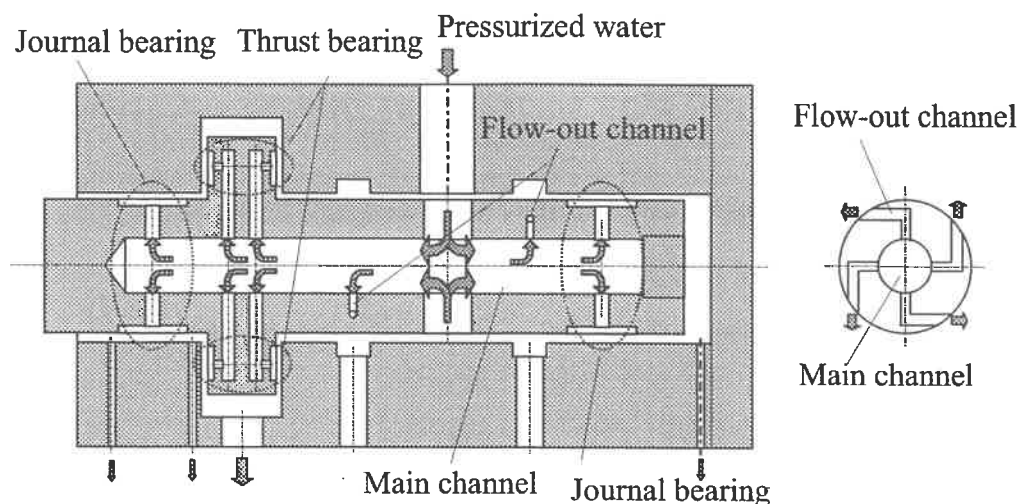


Fig. 1 Structure of proposed spindle

channels; the main channel and the flow-out channels; operate as a built-in motor driven by water flow.

2.2 Supporting function (Water hydrostatic pressure bearing)

Second feature of the spindle is that water hydrostatic bearings support the spindle. The water flow supplied to the spindle is used as lubricant fluid for the water hydrostatic bearings as well as the working fluid for the built-in motor, described in Section 2.1. Advantages of the water hydrostatic bearing of the spindle are the followings:

- (i) Since water is the incompressible fluid, the water hydrostatic bearing is more suitable for the design of an advanced spindle with higher bearing stiffness. In contrast, conventional spindle used for most of the ultra-precision machine tools uses air for hydrostatic bearing as the lubricant fluid. In this case, due to higher compressibility of air, it is not easy to design a hydrostatic bearing with higher stiffness. In spite of the difficulty, an air hydrostatic bearing with higher stiffness is still needed, designer have to design larger bearing surface area and small gap size of bearings. However, larger bearing surface needs large and/or longer spindle. It is apparent that both requirements; i.e., larger bearing surface area and small gap size of the bearing; are very difficult to be attained in machining and fabricating process of the spindle.
- (ii) The viscosity of water is smaller than that of oil, which enables us to operate the spindle with higher rotational speeds.

2.3 Cooling function

Third feature of the spindle is a cooling function for the spindle. Water is effectively cool the spindle, because water flows both inside and outside of the rotating spindle body. Therefore, controlling water temperature minimizes the spindle deformation due to thermal effects. In addition, in comparison with oil, water has an advantage in terms of cooling performance, since the thermal conductivity of water is higher than that of oil.

3. Calculated spindle performance

In order to develop the present spindle in a design process, it is the most important issue to determine sizes of main parts of the spindle that affect spindle performances. For example, design parameters to be

determined are gap size between casing and spindle, diameter of the flow-out channel, gap sizes at each hydrostatic bearing and so on. In this study, mathematical model to represent spindle performance have been derived. Based on the mathematical model, design algorithm and software have also been developed to determine principal design parameters in order to satisfy given specifications; i.e., rated power, rated rotational speed and stiffness of hydrostatic bearings. The developed software for spindle design can calculate 10 design parameters. Using the software, a set of optimum parameters can be calculated so as to obtain highest efficiency at given operational condition. Spindle performances obtained by theoretical studies are given by Figs. 2 to 4, respectively.

4. Experimental results

Figure 5 shows the developed spindle. A performance of rotational speed is given by Fig. 6. This figure shows experimental result as well as theoretical result. It is shown that a rotational speed of the spindle reaches 10,000 rpm by supplying 20 l/min water flow.

Figure 7 shows water temperature during spindle operation. A water temperature control unit is equipped to a water supply unit, and water temperature was set to 20 degree in this experiment. As a consequence, water temperature was controlled from 19.8 to 20.8 degree. Therefore, it is considered that the water flow inside and outside of the spindle body can minimize spindle deformation due to the thermal effects by not only spindle operations but also cutting operations.

5. Summary

A spindle driven by water flow energy for ultra-precision machine tool was presented. Principles of the spindle operation and spindle structure were also presented. Theoretical and experimental studies showed performances of the spindle. Rotational motion accuracy of the spindle and stiffness of the water hydrostatic bearings will be measured in future work. Then the developed spindle will equipped with an ultra-precision machine tool for precision machining experiments.

Motor, hydrostatic bearings and cooling function, which are all operated by water flow energy, are fully integrated into the developed spindle, effectively. In addition, since water is the incompressible fluid, water hydrostatic bearings has an advantage for designing higher stiffness bearing with small bearing size. In fact, the developed spindle size is small. This advantage means that the spindle is suitable for the meso scale precision machining.

The developed spindle can be operated by water flow, including pure water. It is, therefore, considered that the spindle can be used in semi-conductor plants. New applications in such attractive field will be explored.

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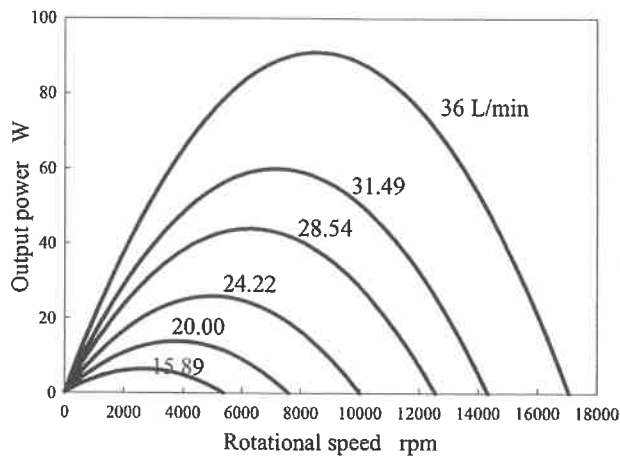


Fig. 2 Relation between rotational speed and power

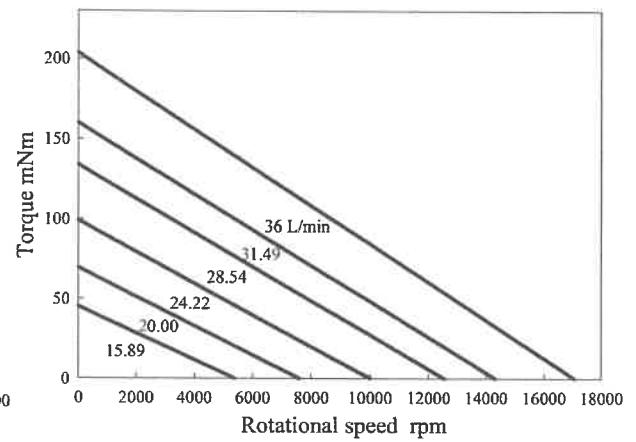


Fig. 3 Relation between rotational speed and torque

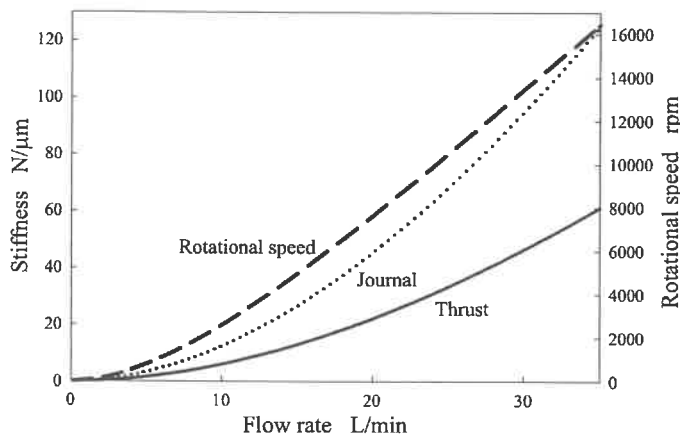


Fig. 4 Relation between flow rate and stiffness and rotational speed

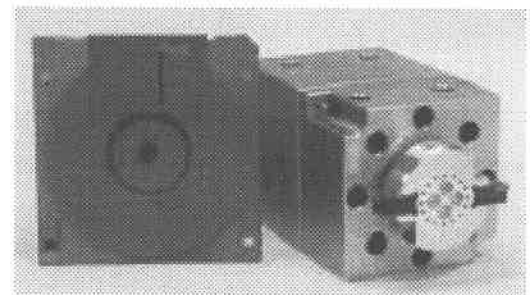


Fig. 5 Developed spindle

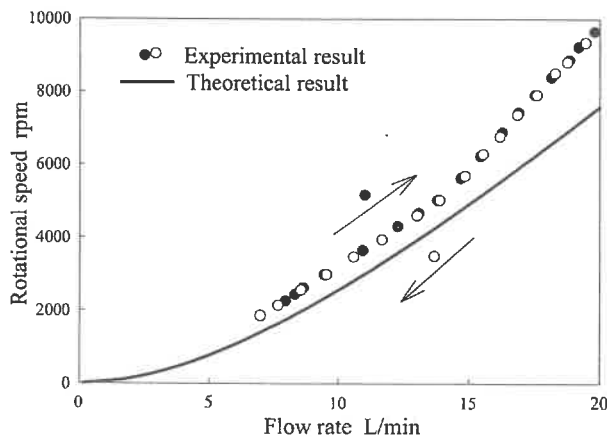


Fig. 6 Relation between flow rate and rotational speed

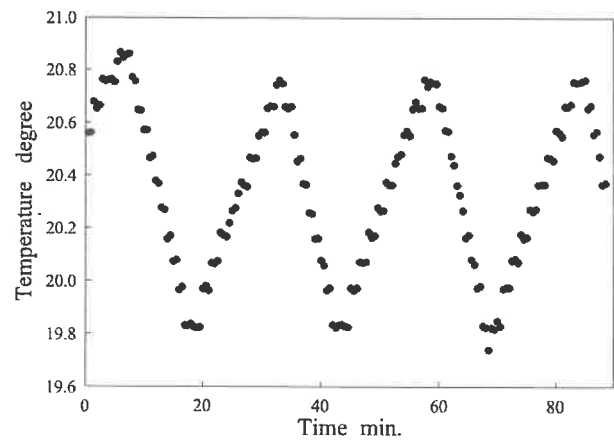


Fig. 7 Cooling water temperature

Design and Characterization of an Aerostatic Spherical Bearing

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Abstract

An aerostatic spherical bearing has been developed for parallel kinematic machines. The design is based on a ball and socket configuration and is preloaded by permanent magnets arranged about the socket. High pressure air is injected through three orifices into the cavity between the ball and socket to produce a low friction, high accuracy interface. Joint accuracy is measured using a coordinate measurement machine and found to be $16\text{ }\mu\text{m}$ throughout the joint working range; this can be reduced to $11\text{ }\mu\text{m}$ through calibration of the joint error parameters. The flow properties are modeled using a Navier-Stokes flat-plate approximation and agree with measurements to 10% within the joint operating range.

1 Introduction

Aerostatic bearings are commonly used in high accuracy applications such as coordinate measurement machines (CMM) and high-speed spindles. Porous materials or discrete orifices are typically used to introduce compressed air into the region between two precision surfaces. For small air gaps, this produces a stable fluidic layer with significantly lower friction than sliding lubricated surfaces. The most common examples of aerostatic bearings are rectangular or circular flat plate bearings that are used to support the linear stages on a CMM; and cylindrical bearings used to support spindle shafts for high speed machining centers [1]. Spherical aerostatic bearings offer opportunities for motion in three dimensions, namely rotary motion about each of the joint's x-, y- and z-axes. A ball and socket configuration represents a high precision design for a spherical joint, as all three

axes are co-located [2]. This reduces the errors inherent in kinematic three degree-of-freedom structures such as modified universal joints, or other mechanical configurations [3].

This paper describes the development and testing of a three degree of freedom aerostatic bearing. Section 2 describes the design and construction of the bearing and Section 3 models its accuracy and fluid flow properties. Finally, in Section 4 the models are compared with measured data from the joint prototype.

2 Design Process

The aerostatic joint is required to have a singularity-free working range of 180° and an accuracy of several micrometers. The joint prototype has an aluminum socket turned from 90 mm cylindrical stock (alloy 6061) and a 50 mm diameter (grade 50) solid steel ball bearing. A magnetic preload between the ball and socket is provided by eight 12 mm diameter magnets in a single ring around the socket.

An accurate mounting surface between the ball and the socket is created through replication with a Teflon-laced epoxy [4]. The ball is coated with a wax compound to achieve an even surface layer for a nominal initial air gap height of approximately $10\text{ }\mu\text{m}$. Additionally, a mold release is applied to the wax coating to facilitate removal of the ball following replication. The resultant surface finish may be rougher than for direct replication of a steel surface because of the wax compound; furthermore, plastic deformation of the wax due to the high contact forces at the orifices may cause slight indentations in the replicated surface. These irregularities can be removed with fine grit sandpaper. High-pressure air is then injected into the base of the socket via three $50\text{ }\mu\text{m}$ diameter ruby orifices [5], creating an air gap that supports the ball.

The replicated socket is shown in Figure 1 and the

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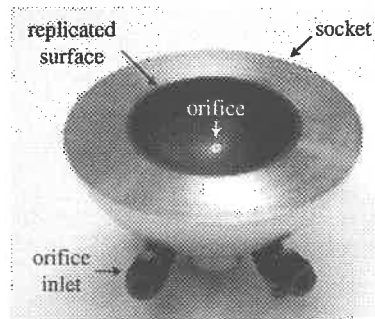


Figure 1: Fluid contact socket.

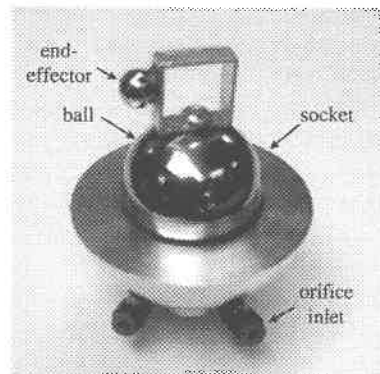


Figure 2: Fluid contact joint.

complete joint with ball and end-effector is shown in Figure 2.

3 Modeling

3.1 Accuracy Model

Measurement and calibration of the joint's positional accuracy is achieved using calibration techniques developed for industrial manipulators [6]. The modeling of the ball and socket joint is presented elsewhere; measurements of similar joint prototypes based on a rolling contact mechanism show a positional accuracy of $12\ \mu\text{m}$ uncalibrated and $8\ \mu\text{m}$ calibrated [7]. Using a Monte Carlo analysis [8], the aerostatic joint accuracy is predicted to be $11\ \mu\text{m}$ by considering errors such as ball asphericity and replication errors.

3.2 Fluid Model

A simplified Navier-Stokes representation is used to model the fluid flow characteristics. Due to the high aspect ratio between joint surface area (several square centimeters) and air gap (tens of micrometers), the joint is approximated by two parallel flat plates. Given the orifice specifications – pressure drop

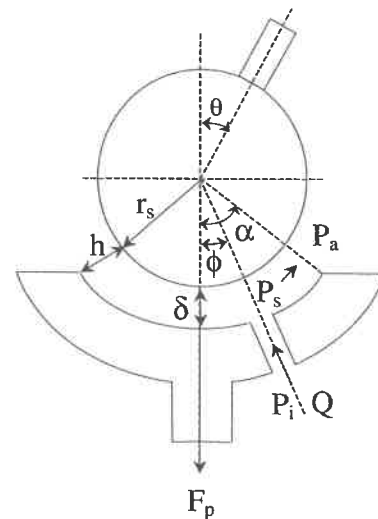


Figure 3: Socket flow schematic. Note that the schematic is not to scale and $r_s \gg h$.

versus flow rate characteristics – the inlet pressure can be expressed as a function of the air gap. Measurement of the air gap versus inlet pressure is then used to validate the flat-plate approximation.

The maximum preload force for the aerostatic joint is measured to be $90\ \text{N}$ for the single row of magnets. The volume flow rate and pressure required to create an air gap under this preload force needs to be determined for a given orifice aperture diameter. The pressure and flow schematic is shown in Figure 3, where the orifice air supply enters at gauge pressure P_i and flow rate Q , at an angle ϕ from the vertical and exits at the socket edge (angle α) with atmospheric pressure P_a . The flow rate is constant through the air gap due to the assumption of incompressible flow. The pressure drop P_s along the socket supports the ball, hence the total vertical component of the pressure force must support the bearing against the magnetic preload force F_p . This is modeled as the ratio of the preload force to the projected socket area:

$$P_s = \frac{F_p}{\pi (r_s \sin \alpha)^2} \quad (1)$$

The Navier-Stokes equation in the radial direction can be simplified and solved to determine the pressure drop across the orifices:

$$P_o = \frac{6\mu P_s R_o}{\pi h^3 N_o} \ln \left(\frac{l_s}{l_o} \right) \quad (2)$$

Where the socket arc length l_s is the distance swept by the socket angle, scaled by the plate-sphere radius ratio η :

$$l_o = \eta r_s \alpha \quad (3)$$

The orifice arc length l_o is calculated in a similar fashion. The scaling factor η is determined by equating the surface area of a sphere of arc angle α and radius r_s ($\frac{\alpha}{\pi} \cdot 4\pi r_s^2$), to the equivalent area of a flat-plate of radius r_p (πr_p^2), leading to the relationship:

$$\eta = \frac{r_p}{r_s} = \sqrt{\frac{4\alpha}{\pi}} \quad (4)$$

The resistance of the orifice R_o is supplied by the manufacturer, N_o is the number of orifices used and μ is the viscosity of air. The inlet gauge pressure P_i is the sum of the socket and orifice pressure drops:

$$P_i = P_s + P_o \quad (5)$$

The socket outlet is believed to act as a restrictor for the fluidic system. The air gap h is therefore assumed to be the separation of ball and socket at the socket edge and is geometrically related to the vertical displacement δ :

$$h = \delta \cos \alpha \quad (6)$$

For socket angles approaching 90° this relationship degrades and an average air gap should be calculated using the cubic weighted average of the gap heights along the socket. The relationship between the inlet pressure P_i and the vertical displacement δ is compared to measured data in Section 3.2 to validate this simplified Navier-Stokes approach.

4 Measurements

4.1 Accuracy Measurements

The maximum and mean errors for the aerostatic joint are shown in Figure 4, showing a maximum uncalibrated (geometric) error of $16 \mu m$ and a calibrated (kinematic) error of $11 \mu m$. This compares favorably with the predicted accuracy of $11 \mu m$. The large maximum error is believed to be due to slight inaccuracies in the replication process. The uncalibrated mean error is found to be $4 \mu m$ which renders this joint suitable for high accuracy robotic applications.

4.2 Fluid Measurements

A coordinate measurement machine accurate to $2 \mu m$ is used to measure the displacement of the ball center as the inlet gauge pressure is increased from 0 to $7 atm$ as shown in Figure 5. For inlet pressures above $3 atm$,

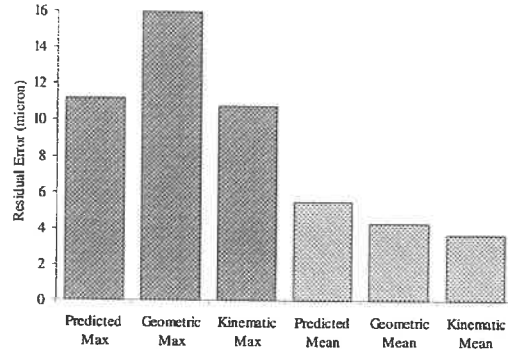


Figure 4: Comparison of joint accuracy.

corresponding to an air gap of about $20 \mu m$, the measured and predicted vertical displacements agree to within 10%. However, at lower pressures the model underestimates the displacement and is accurate only to 15%, believed to be due to surface irregularities affecting fluid flow at small gap sizes. For inlet pressures greater than $7 atm$ pneumatic hammer is observed, which is a common occurrence for large air gaps [9]. Therefore, for a desired operating pressure range of 3 to $7 atm$, the flat-plate Navier Stokes model represents a suitable first order approximation.

To improve the correlation between predicted and measured displacements the Navier Stokes in spherical coordinates could be used. Alternatively, the outlet gap size h could be directly measured with capacitance probes and compared to the predicted. The volume flow rate Q can also be measured with a flow meter and compared to the predicted flow rate.

5 Conclusion

The spherical aerostatic contact joint is approximated by a first order flat plate model in order to predict the pressure versus displacement characteristics. Within the inlet pressure operating region of 3 to $7 atm$, corresponding to an operating air gap of $20 \mu m$, the model agrees with measurements to within 10%. This is sufficient to allow first order predictions of the aerostatic contact joint performance. The agreement between measured and predicted vertical displacements for pressures below $3 atm$ is only to within 15%, assumed to be due to the poor replication surface finish. Direct replication of the ball could improve the surface finish and consequently the low-pressure performance.

The joint accuracy maximum error is $16 \mu m$ uncalibrated and $11 \mu m$ when calibrated. This is slightly higher than predicted, due to the inaccuracies inher-

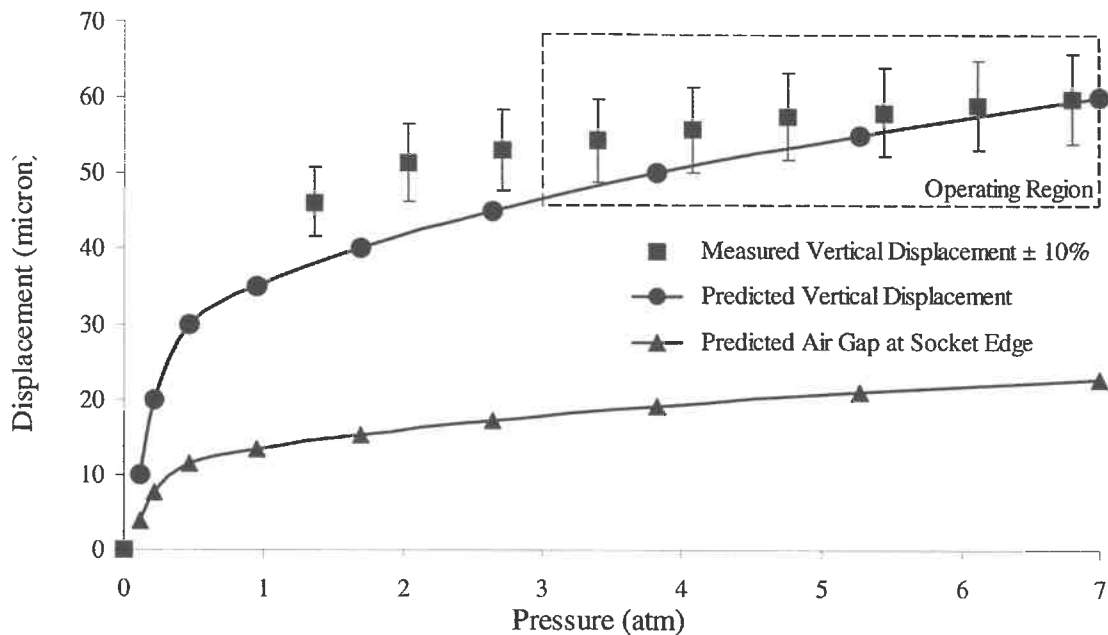


Figure 5: Pressure-displacement characteristics for aerostatic contact joint. The predicted and measured vertical displacements agree to within 10 % for the inlet pressure operating region of 3 to 7 atm.

ent in the replication process. Improved control of the replication process and resultant surface finish should result in a smaller operating gap and an improved overall accuracy.

Additional work is required to characterize the joint stiffness and damping. Preliminary tests indicate that the aerostatic joint stiffness is approximately $5 \text{ N}/\mu\text{m}$ which agrees to within 10 % with the predicted stiffness from the Navier Stokes flat-plate model. Damping is noticeable in the aerostatic joint, believed to be due to the viscosity of air in a small gap and hysteresis from induced magnetic fields in the ball. Characterization of the joint damping and stiffness would allow a simple spring-mass-damper model to be developed and the dynamic performance evaluated.

6 Acknowledgements

The authors would like to thank ABB for their support during this research.

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Keywords: aerostatic bearing, spherical joint, parallel kinematic machines

Theoretical and Experimental Determination of the Stiffness Properties of a Capstan Drive

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Wire capstan drives are used as rotary transmission elements for their very low (nominally zero) backlash and high stiffness properties. To obtain high stiffness, the cable is typically wrapped around the input and output drum in a figure-eight pattern multiple times. This stiffness can be determined by analyzing the amount of deformation between the cable and the drums and the friction caused by this contact. In product literature and various textbooks [1][2][3], it was noticed that different methods of determining the stiffness of the transmission drive were presented depending how friction was considered. It is found the model which considers the extension from the external torque and preload force matches experimental data along with the other model that defined the area of traction to act in a certain section of the drum. As the coefficient of friction between the cable and transmission drum increases, the two models differ giving the method that defined the area of slip along a certain section of the drum, the conservative choice. This paper presents a comparison of two different stiffness models, along with data from experiments to study the model differences.

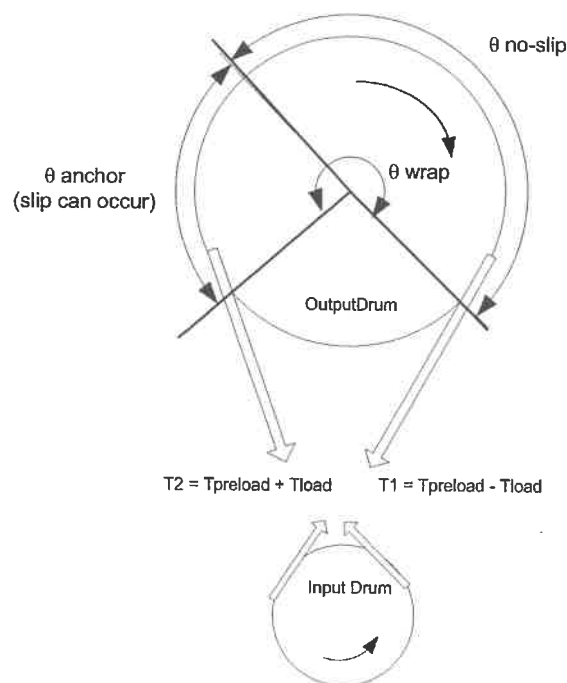


Figure 1: Definition of “angle-of-anchor” and “angle-of-wrap” for the output drum rotating counterclockwise.

The main difference discovered between the two stiffness models is due to how the traction region of the cable is defined. Traction can be defined as acting over the angle-of-anchor which is the section of the cable where slippage can occur while the cable is loaded. This angle is the angle one would derive from the classic capstan equation as being required to hold the varying load, given a cable preload. Alternatively, traction can be defined as acting over the angle-of-wrap which considers both sections of the cable where slip and no-slip occur. These two angles and the area of slip and no-slip are shown in Figure 1. If traction is defined by the angle-of-anchor, the section of slippage is due to the external torque applied. For traction defined by the angle-of-wrap, extension due to the preload force through the cable contacting the drum must also be considered in addition to the external torque that the angle-of-anchor considers. Therefore, half of the physical behavior of the cable is considered in the angle-of-anchor where the complete behavior of the cable is considered in the angle-of-wrap.

To determine which model was correct, experiments were conducted to measure rotary stiffness. The test setup consisted of a 3 mm steel cable wrapped in a figure-eight pattern around two transmission drums, with diameters 50 mm and 280 mm, refer to Figure 2. A load cell was spliced in the cable to determine the force around the drum and a rotary encoder attached to the output transmission drum determined the rotation of the output shaft, refer to Figure 3. A lever arm attached to the input transmission drum determined angular deflection of the input shaft. This value was subtracted to get the net deflection of the cable. A separate measurement which involved tensile testing with an Instron machine was performed to determine the effective modulus of the 3 mm 7X19 stranded steel cable.

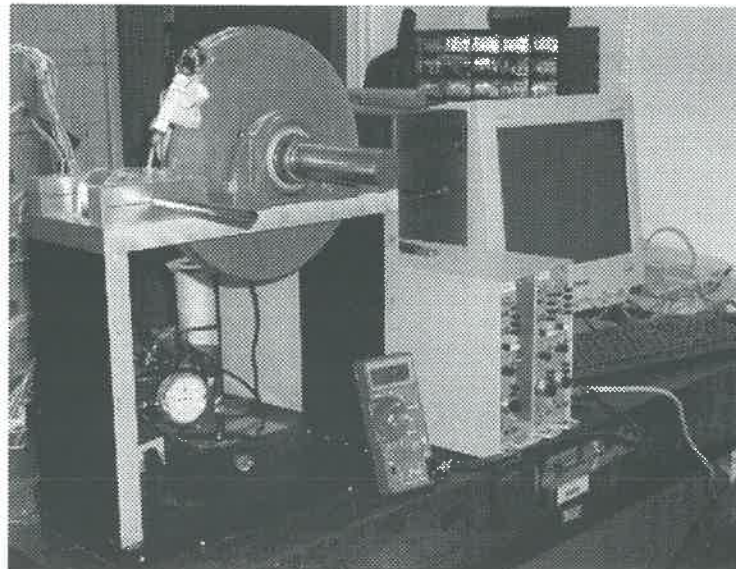


Figure 2: Configuration of the Capstan experimentation with load cell spliced between the cable.

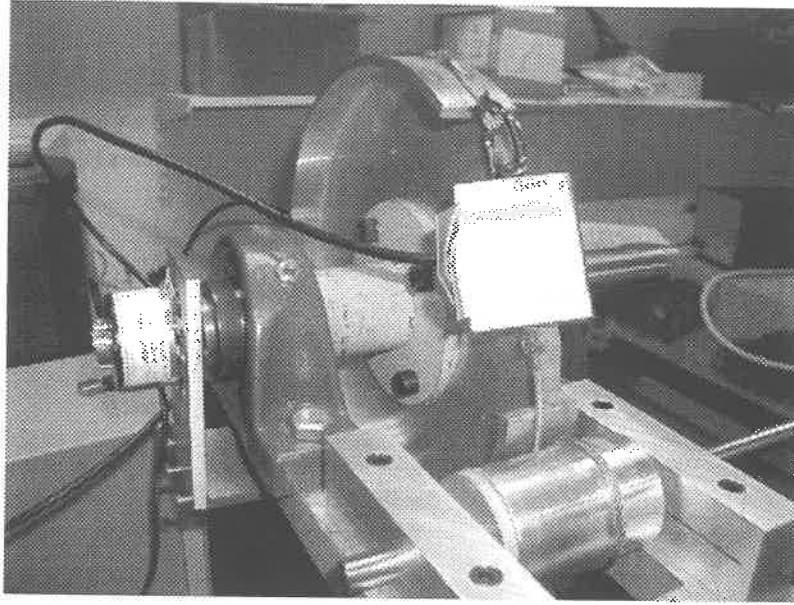


Figure 3: Encoder and load cell on output transmission drum. Input transmission drum shown in the bottom right.

Based on these experiments, it was determined that both models predict the correct torsional stiffness, varying 6% given an angle-of-anchor of 120°. When compared to experimental values, 60% of the results fell within both models, as shown in Figure 4. Likewise as the angle of anchor is increased, both models predict the same torsional stiffness within 6%. Therefore for low coefficient of friction the angle-of-anchor or the angle-of-wrap can be used where the cable extension is due to the external torque and preload in the cable but when the coefficient of friction increases and even though the angle-of-anchor is small, the angle-of-anchor is the conservative estimate. The recommended conservative torsional stiffness equations for a single free-set of cables are:

$$K_{torsional} = \frac{KD_o^2}{4} \quad (1)$$

$$K = \frac{2AE}{2 \frac{D_o + D_i}{\mu} + L_{free}} \quad (2)$$

Given the maximum preload tension, $T_{preload}/2$, is used in the cable which makes the maximum torque:

$$\Gamma = \frac{T_{preload} D_i}{4} \quad (3)$$

where AE is the effective modulus of the cable area, D_O and D_I are the output and input drums respectively, L_{free} is the length of the cable not in contact with the drum, and the coefficient of friction between the drum and cable is μ .

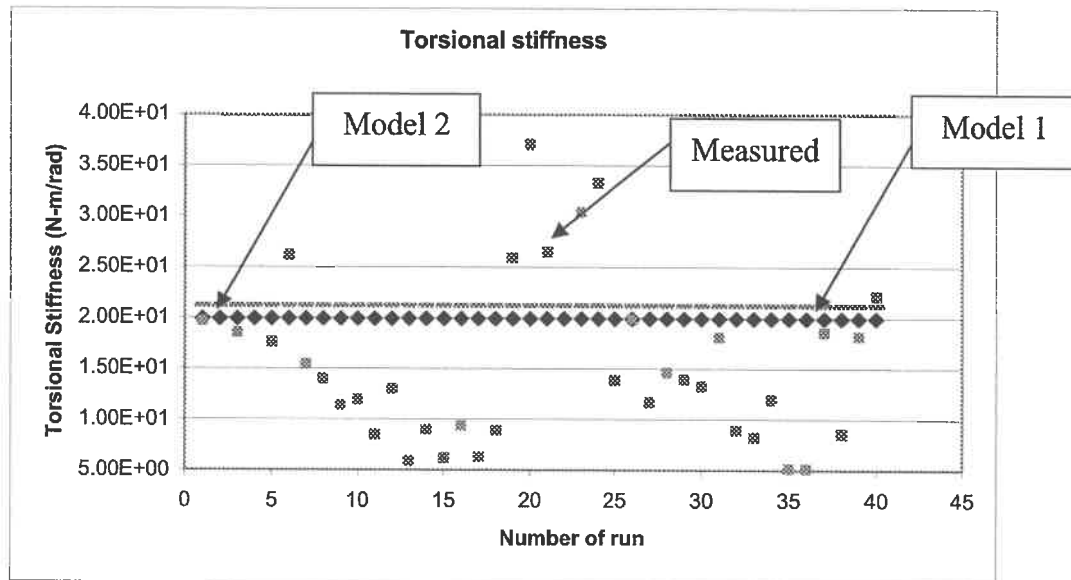


Figure 4: Torsional stiffness for measured and calculated values¹.

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¹ Note two data points with values of 70 and 100 N-m/rad, are not shown on the graph due to randomness in value.

Experimental Determination of Kinematic Coupling Repeatability in Industrial and Laboratory Conditions

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While kinematic couplings are frequently used to create repeatable interfaces in ideal environments, their use in typical industrial interfaces is less frequent, as previous studies have described couplings mostly under modest loads and controlled environmental conditions. On the other hand, the performance of heavily loaded industrial equipment, such as robots used in automotive production, could benefit from highly repeatable interfaces for assembly and replacement of robot modules. A series of tests was conducted to assess the repeatability of large kinematic couplings on the ABB 6400R industrial robot, shown in Figure 1. Two of the more frequently replaced interfaces, the robot base to factory floor interface and the robot wrist interface, were used as test locations. While the base interface represents a high load, large size coupling, the wrist interface requires a smaller coupling subjected to a moderate load.

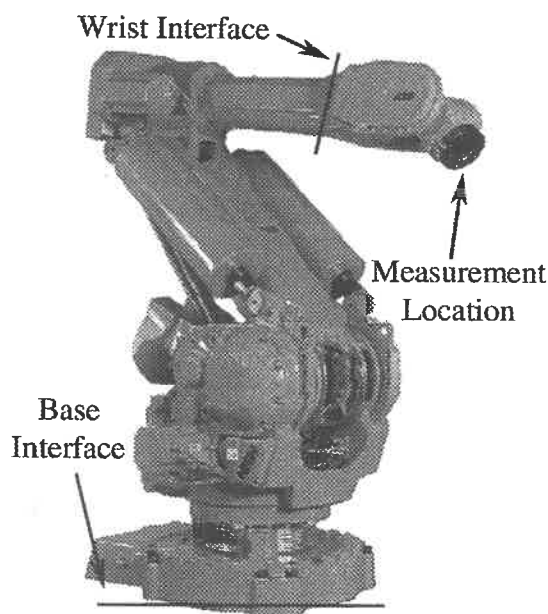


Figure 1 - ABB IRB 6400R Industrial Robot

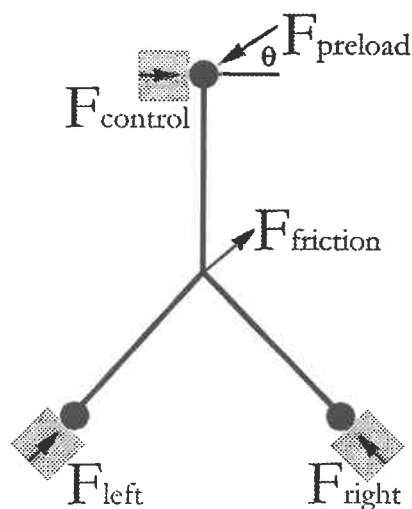


Figure 2 – Force Schematic for Three Pin Coupling

For both interfaces, replacements for the existing pin joint interfaces were designed using two forms of kinematic couplings – the three-pin coupling and the canoe-ball coupling. The three-pin coupling replaces the non-deterministic pin constraints with three line constraints for the in-plane degrees of freedom and a large planar constraint for the remaining degrees of freedom. Figure 2 shows a force schematic for the three pin coupling with the three pins shown in black and the mating surfaces in light grey. The canoe-ball couplings employ large radius spherical surfaces ground onto compact elements with matching groove elements to constrain the interfaces using six contact patches. Figure 3(a) shows a single canoe ball element with its mating groove element. To maximize stability, each coupling element is placed in the

configuration shown in Figure 3(b) where the centerline of each groove bisect the coupling triangle and intersect at the coupling centroid.

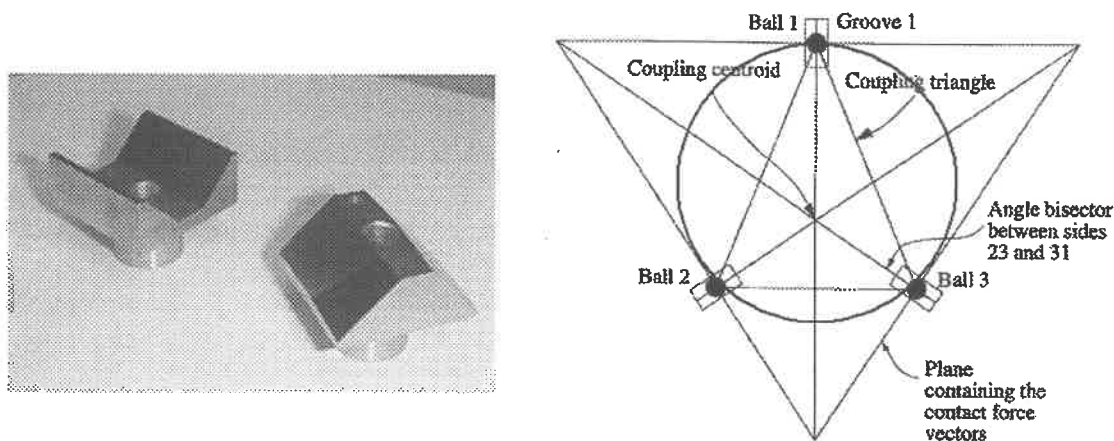


Figure 3 – (a) Canoe Ball and Groove Elements and (b) Standard Kinematic Coupling Circle

During industrial testing, measurements were taken for an embodiment of each coupling type on the base and wrist interfaces. For both interfaces, separate coupling plates containing the contacting elements were attached to the existing interfaces. The Leica LTD500 laser-tracking measurement system was used to measure the location of a retroreflector mounted on the robot's tool flange. The LTD500 has an accuracy of 0.01 mm per meter between the retroreflector and the motorized laser-tracking device. Each measurement set included a replacement using a basic installation method and one using a refined procedure consisting of the following steps:

1. Remove preload bolts and separate coupling interfaces using an overhead crane for support.
2. Inspect and carefully clean the contacting surfaces with methanol. Surfaces are not lubricated during the experiments.
3. Clean and grease all bolts.
4. Engage coupling interfaces with special attention to alignment of elements using an overhead crane for support.
5. Replace preload bolts and any additional bolts. Tighten bolts to design specification in a stepped fashion following a consistent pattern.

Additional low load, laboratory measurements were taken for the wrist couplings under gravity loading as a baseline comparison to previous studies. Similar deviations in installation procedure were analyzed for the laboratory measurements.

The primary conclusion from coupling performance measurements in both industrial and laboratory settings is that proper mounting of a kinematic coupling is equally important as, if not more important than, the presence of the kinematic interface. Careful attention to mounting conditions is required if kinematic couplings are to provide high repeatability for high-load bearing industrial interfaces. A factor of two to three improvement in repeatability is possible simply by switching from a basic mounting procedure to the refined installation procedure. Without properly applied, sufficient preload, frictional effects determine the interface position and prevent optimal repeatability, although Schouten has noted that flexures can be used to

mitigate this effect. Laboratory measurements provided best case canoe-ball coupling repeatability on the order of $0.1\mu\text{m}$ and three pin coupling repeatability of $1\mu\text{m}$. In industrial trials, the canoe-ball coupling was able to improve repeatability of the base interface by 93% and the wrist interface by 59%. The three-pin coupling was able to improve the base interface repeatability by 93%. Because of damage to the alignment features, the three-pin coupling on the wrist interface must undergo further industrial measurements to properly characterize repeatability. Repeatability data for industrial measurements are summarized in Figure 4 and Figure 5, while numerical data is shown in Table 1. The trends in the measurements indicate that the three-pin coupling is a lower cost, less repeatable alternative to the higher cost, more repeatable canoe ball couplings; however, as the physical size and scale of loading increase, this trend becomes less significant. While measurements for the laboratory experiments were taken at the center of stiffness, measurements for the industrial couplings were taken at a fixed distance from the coupling center. A simple approximation for the repeatability of the industrial couplings at the center of stiffness of each coupling indicates that the theoretical repeatability could be an order of magnitude less than the measured results. In general, both the three-pin and canoe-ball couplings measured in this study result in an order of magnitude improvement in coupling repeatability over the current interface designs for the IRB6400.

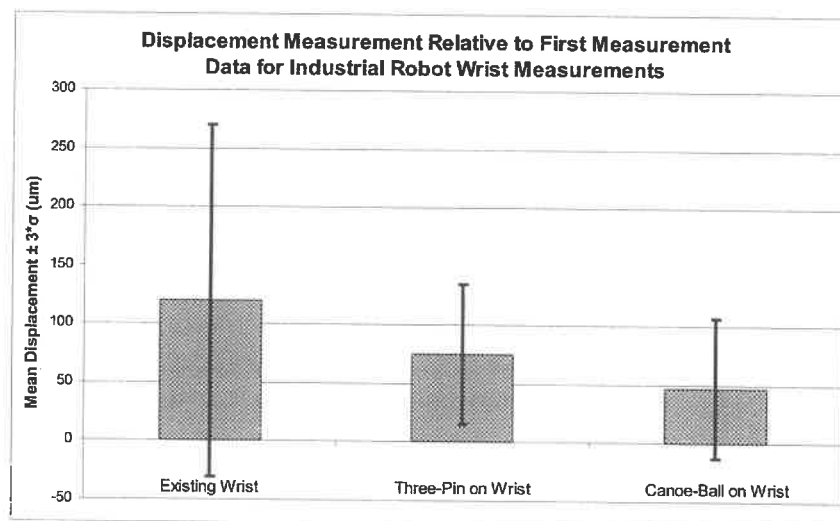


Figure 4 - Summary Plot of Industrial Data for Robot Wrist

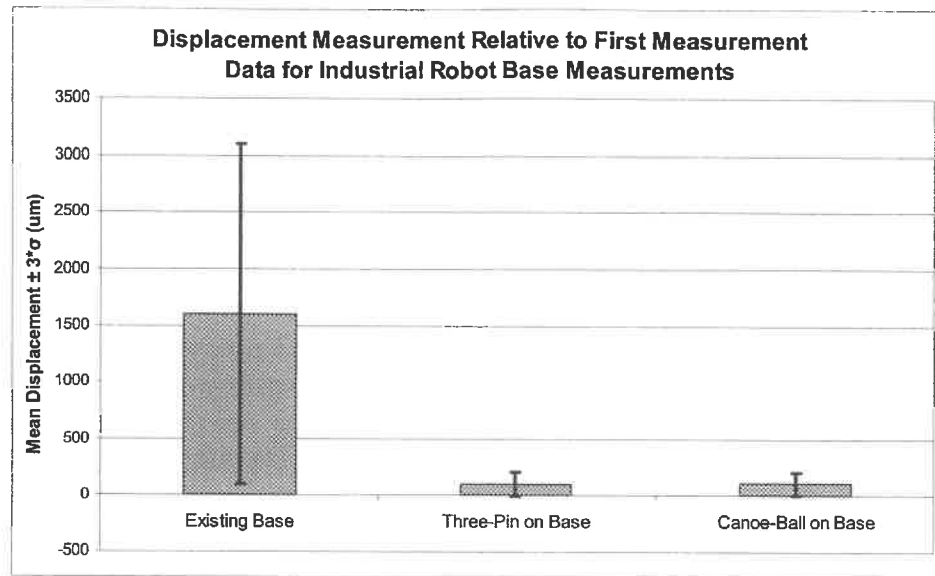


Figure 5 - Summary Plot of Industrial Data for Robot Base

Experiment	Average Location Shift	Standard Deviation
Laboratory Canoe Balls - Unloaded	-0.04 μm	0.14 μm
Laboratory Canoe Balls - Design Preload	-0.69 μm	0.41 μm
Laboratory Three-pin - Design Preload	-0.09 μm	1.01 μm
Industrial Robot Wrist - Canoe Balls	50 μm	25 μm
Industrial Robot Wrist - Three-pin	80 μm	30 μm
Industrial Robot Base - Canoe Balls	110 μm	35 μm
Industrial Robot Base - Three-pin	100 μm	35 μm

Key words: kinematic couplings, robotics, repeatability, precision fixturing, three-pin, canoe-ball

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DESIGN AND DEVELOPMENT OF A HIGH PRECISION LENS FOCUSING MECHANISM USING FLEXURE BEARINGS

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Abstract

The emergence of multi-tier stacked die architecture in semiconductor devices has given rise to the need for multi-level interconnection between dies (bare IC chips) and the substrate. In the context of interconnection by wire bonding, this necessitates the use of a vision system, which has real time focussing capability. This paper presents the design and development of such a high precision Programmable Focussing Mechanism (PFM), which is used to move a lens relative to another fixed lens, thus altering the focus of the optical system. The device employs flexure bearings and is actuated by a voice coil motor (VCM).

Keywords: Machine vision, lens focus, wire bonder, flexure bearing, voice coil motor

1 Introduction

Wire bonding is by far, the most flexible and the most widely used semiconductor interconnection technology [1]. Fine gold wire is used to make connections between the "die" (bare IC chip) and its substrate, using a thermo-sonic bonding process. Modern-day wire-bonders are expected to bond on pads with a tight pitch of around 50 μm or less, with an accuracy of $\pm 4 \mu\text{m}$ at their best. Status of art machines can attain bonding rates of 10-15 wires per second. The bond-head, which carries the ultrasonic transducer, is mounted on a X-Y table, which moves in a horizontal plane at peak accelerations of about 120 m/s^2 in each direction, possibly simultaneously.

Before the onset of bonding every device, a computerised machine vision system precisely locates the positions and orientations of the substrate and the die mounted onto it. This involves locating the leads on the substrate and bond pads on the die and then transforming and correcting the taught locations for each bond. The vision system is also used for post-bond inspection in monitoring bond placement accuracy and wire sweep.

2 Programmable Focussing Mechanism (PFM)

Traditionally, a wire bonder is equipped with a fixed focus vision system of two different magnification levels, high magnification for the die pads and low magnification for the substrate leads. The incessant push for miniaturisation of ICs has resulted in a stacked die architecture, wherein one die is bonded on a substrate and then one or more dies are mounted successively one above the other in a vertical stack. Wire bonding on such multi-tier die stacks needs a vision system with programmable focussing capability for die surfaces at different heights. This is achieved by motorising a relay lens in the high magnification optics [2].

The usable travel range of the PFM is 3 mm with the typical stroke response requirement specified as 0.5 mm in 30 ms with a position repeatability of 10 μm . Although absolute de-centration of 50 μm is allowable between the moving lens and fixed lens, the repeatability in image position and by implication, the repeatability in off-axis deviation of the moving lens is limited to a stringent 1 micron. The tilt excursion of the moving lens about any axis normal to motion has to be less than 12 arc minutes. The maximum "ON"-time during image capture on the largest of dies is specified as 5 seconds and a maximum process duty of 10 %.

Fig. 1a shows a schematic diagram of the PFM. The moving lens is affixed in a cylindrical tube, which is suspended on two flexure disc assemblies. The space between the flexures is utilized to house a small voice coil motor, which actuates the moving lens and precisely controls its axial position in closed loop servo mode using feedback provided by an LVDT. The position of the fixed lens can be adjusted and locked as and when required for different batches of packages to be bonded. This extends the overall focus range of the optics to the required 3mm, while keeping the stroke requirement of the moving lens to within $\pm 0.7 \text{ mm}$ for any given package, leading to a compact design.

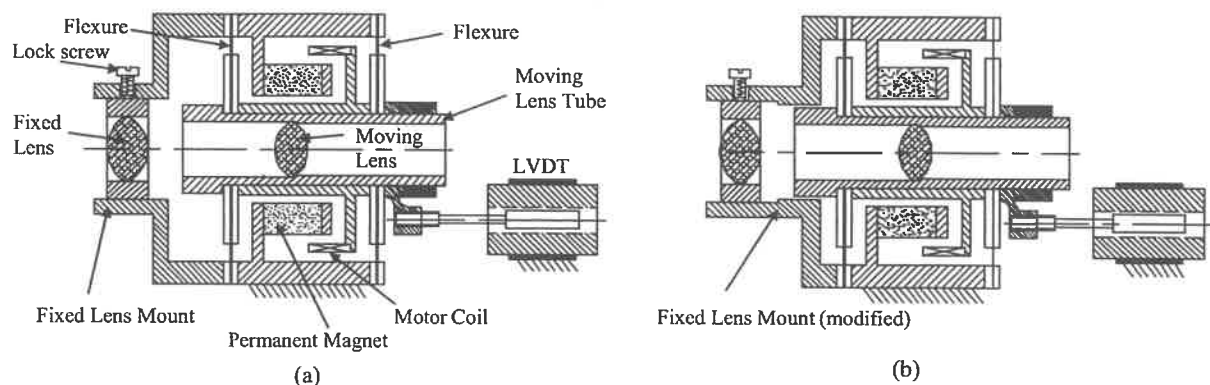


Fig. 1 Schematic diagrams of the Programmable Focusing Mechanism; (a) un-damped, (b) with integral air damper

Fig 1b shows a slightly modified version of the PFM in which the design of the fixed lens mount has been altered to surround the moving lens tube over a significant portion of its length such that a small radial clearance ($\sim 10\text{-}15\text{ }\mu\text{m}$) exists between the two. This simple modification introduces substantial air damping in the system when air is driven through the annular gap, to and from the space between two lenses. The objective of this damping is to attenuate the residual vibrations at the end of the stroke, thus reducing the settling time and speeding up the process.

3 Design of flexure bearings

Stringent demands on the repeatability of lens position and orientation, call for the use of flexure bearings. Flexures offer incomparably high repeatability of motion since motion is enabled through elastic deformation of the material. Absence of relative motion between two contacting parts leads to frictionless, wear-free operation. Thus, if designed adequately to withstand fatigue, flexure bearings can easily outlast rolling element bearings and slider bearings. They are relatively inexpensive to produce and simple to assemble.

Fig. 2 shows elements and assembly of the flexure bearing. A flexure disc is photo-chemically etched out of a flat metallic sheet to give three flex-arms placed 120° apart. These flex-arms are partly straight and partly arc-shaped to best utilise the available space while minimising the axial stiffness. Boundary constraints for the flexures are defined by sandwiching them between spiders in the central moving portion and peripheral spacers on the outer section, which is stationary. As shown in Fig. 1, a pair of such flexure discs separated by a distance, forms the bearing.

3.1 Finite Element Analysis of the flexures

Simple analytical methods are inadequate in predicting the operating characteristics of the flexure bearing due to the mutual coupling effect between the three flexural arms making up the bearing. Use of finite element analysis (FEA) method in tackling such a geometrically non-linear problem thus becomes indispensable. FEA has been conducted with a view to examine, apart from the stress distribution, characteristics such as axial stiffness, radial stiffness and extent of parasitic rotation as a function of axial displacement. Fig. 3 shows the FEA results for a single flexure.

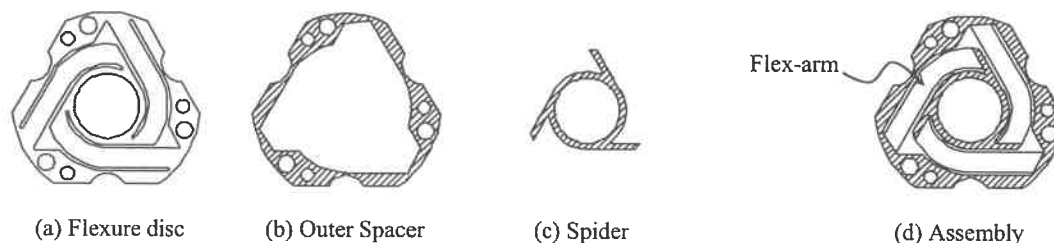


Fig 2. Flexure bearing components and assembly

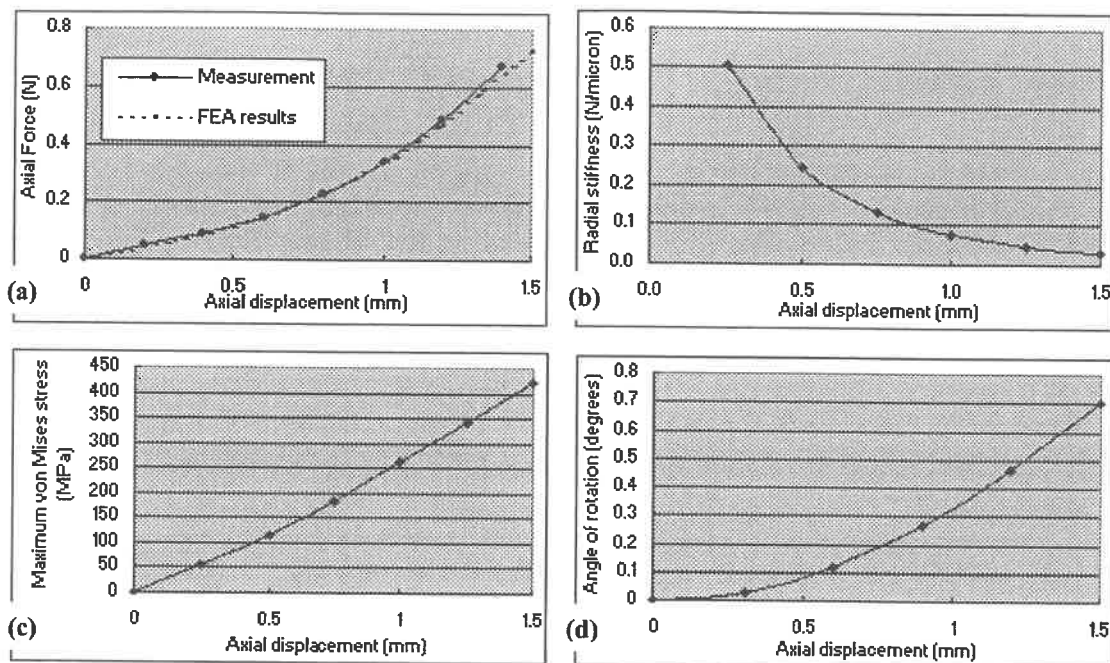


Fig. 3 FEA results for single flexure disc; (a) Axial force (b) Radial stiffness (c) Combined Stress (d) Parasitic rotation

As shown in Fig. 3a, the measured values of **axial force** corroborate well with those predicted by FEA. The restoring force due to the flexure bearing (a pair of flexure discs) for 0.7 mm axial displacement is about 0.4 N. This maximum axial force exerted by the flexure bearing, forms an input to motor design.

Radial stiffness of the flexure bearing (Fig. 3b) reduces rapidly as the flexures move out of plane. The maximum lateral force on the flexures is the inertial force on account of the high acceleration of the XY table, perpendicular to the flexure axis. In the present case, the suspended mass is only about 12 gm. At a maximum acceleration of 120 m/s^2 , the inertial force is 1.44 N. For a stroke of 0.7 mm, the effective radial stiffness of the bearing is about 0.31 N/ μm . Thus the worst-case lateral deflection of the bearing is about 4.6 μm . However this deflection is fully recovered as soon as the table comes to rest, since the equivalent stress is well within the elastic limit.

For a stroke of 0.7 mm, the maximum von Mises **stress** under combined axial and lateral loading is about 171 MPa, Since the maximum stress level is well below the endurance limit of the material, the flexures will have a virtually unlimited life even for completely reversed loading. This has been borne out in the accelerated reliability tests conducted on eight flexure discs which could sustain over 200 million reversed cycles without failure, even at $\pm 1.5 \text{ mm}$ (double the required stroke) giving strong confidence as regards the fatigue life of the flexure bearing.

Deflection of a simple cantilever flexure through a certain distance is always associated with a second order **parasitic error** in an orthogonal direction towards the fixed end of the flexure. In the present design, the parasitic errors of all the three flexures are so coupled that the resultant error motion is a small rotation about the translation axis. Fig.3d shows the variation of this angular motion with axial displacement of the lens tube. The rotation being very repeatable and only a small fraction of a degree, does not have any adverse impact on the imaging process since the moving optical component is an axi-symmetric lens.

4 Voice Coil Motor (VCM) design

Since the travel range of the PFM is only $\pm 0.7 \text{ mm}$, a voice coil motor of cylindrical topology has been adopted. This consists of an axially magnetised NdFeB ring-magnet of high energy density (48 MGOe) and soft iron poles forming a working gap with a radial magnetic field. A copper coil suspended in the working gap (see Fig. 1) develops the axial actuating force, the direction of which depends on the direction of coil current.

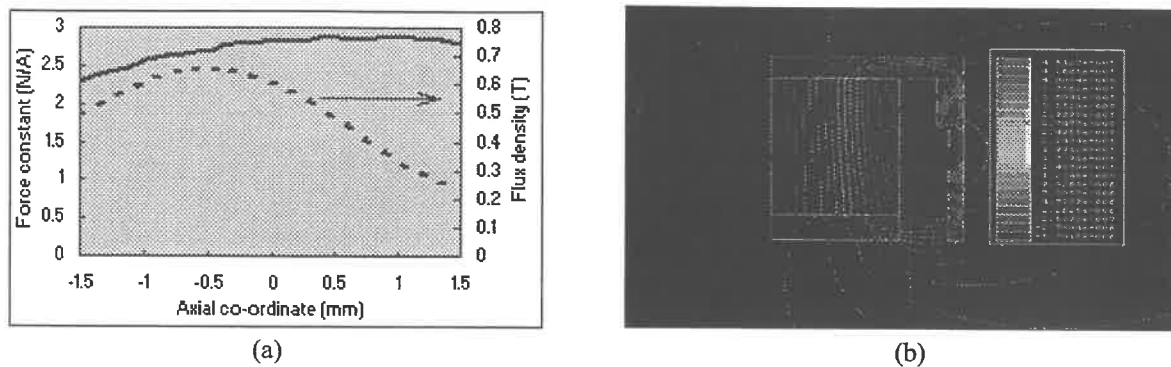


Fig. 4 Electromagnetic FEA results, (a) Gap flux density and force constant variation and (b) flux distribution

4.1 Electromagnetic FEA

The VCM has been optimised using electromagnetic FEA, the results of which are summarised in Fig.4. The maximum flux density of 0.65 T occurs at a location about 0.5 mm from the middle towards the outer end of the working gap. However, the force constant increases from 2.3 N/A at the outer extreme of the stroke to a maximum of about 2.9 N/A close to the inner end. This is because as the coil moves inside the gap, it can also utilise the substantial leakage flux emanating from the magnet as seen in Fig. 4b. In the region of interest, i.e. ± 0.7 mm, the variation in force constant is only about $\pm 4\%$.

Assuming a conservative average force constant of 2.3 N/A, the maximum holding current was expected to be 0.17 A. In actual practice, the current was measured to be 0.2 A and the corresponding temperature-rise in the coil was less than 1°C for the specified maximum "ON"-time of 5 s. The worst-case r.m.s. force is a time weighted average of the maximum flexure force and the inertial force in the axial direction. For a 10% duty cycle, the estimated worst-case r.m.s. force is only about 0.09 N. Hence the thermal test is sufficiently conservative and safe.

5 Conclusion

The combination of voice coil motor and flexure bearing has resulted in a very compact and robust motion stage for the lens focussing application, giving very high position repeatability and overall reliability than would be possible using conventional rolling element bearings and/or rotary-to-linear actuator. Tests aimed at evaluating the dynamic performance have been conducted on several prototypes of the PFM, using a dedicated, auto-tunable FPGA-based controller board [2]. The lens can be moved from its home position to either end of the stroke range of ± 0.7 mm in less than 14 ms, well within the required specification. Imaging tests have also yielded satisfactory results [2]. Patents have been filed in several countries [3]. The PFM has been successfully deployed in the field.

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Synthesis of the 3D Linux-Based CNC System for Precision Machines

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Abstract

This paper proposes a design methodology for a Linux based CNC control system with open architecture capabilities. The developed Linux-based CNC system consists of several software modules such as NCK, PLC, and MMI modules as well as I/O devices. These modules and basic I/O devices are integrated on a single processor platform. In order to verify the performance of the developed system, linear and circular motion errors are measured according to operating conditions in a precision x-y positioning system. In addition, the developed CNC system is applied to a horizontal arm-type CMM with a jerk-limited linear interpolator.

Keywords: Contour Error, Jerk-limited Linear Interpolator, Linux system, NCK, Open-Architecture Controller, PC based CNC, Real-Time Kernel

1. Introduction

Most CNC systems have been developed as a closed architecture and supported by the proprietary technology. Owing to the lack of openness in controllers used today, developers are commonly confronted with closed and proprietary CNC functions. It is virtually impossible to incorporate different types of controllers and sensor-based control schemes [1,2]. Therefore, a Linux based CNC control system with open architecture capabilities is studied to reduce the limitations in this paper.

The Linux based CNC kernel consists of several software modules on a single processor platform so that particular kernel modules can be easily added to and/or removed from the Linux kernel dynamically. The developed CNC system can be integrated with several other functions so as to realize an intelligent CNC system.

Two control schemes are used in the developed Linux-based CNC system. The first control scheme is a cross-coupling control (CCC) scheme to reduce contour error in two-dimensional motion [3]. The second one is a linear interpolator which employs jerk-limited acceleration profiles for preventing structural vibrations and increasing positioning accuracy in a horizontal arm type coordinate measuring machine [4].

2. Architectures of Linux-based CNC system

A Linux-based CNC system based on the software

PC-NC [1] is designed to utilize the PC hardware as shown in Fig. 1. Many advantages of PC systems can be incorporated in the developed system. The real-time task is executed in modules potted in the real-time Linux kernel. Many tasks performing various control functions are constructed for the Linux-based CNC system.

3. Design of CNC module

Since Linux and RT-Linux operating systems have open source codes, small-occupied memory size of the real-time kernels, stability and low latency characteristics, as well as free availability, a CNC platform is constructed by using them. Real-time tasks may have higher priorities compared to the Linux kernel, and may preempt it in order to meet deadline requirement.[5] A GTK+ language is used to develop MMI modules. Various control modules are written in C-language.

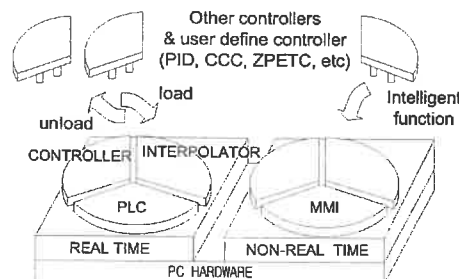


Fig. 1 Architecture of Linux-based CNC system.

A. Real-time modules

In this paper, real-time task is executed in modules potted in the real-time Linux kernel. Real-time modules require correctness not only in computing results but also in time responses. The NCK (Numerical Control Kernel) and PLC modules are programmed in RT-Linux and implemented by the multi-thread technology. These modules can be loaded into the memory dynamically.

The NCK of Linux-based CNC system consists of three major parts: a 3D motion command generation part, a motion control part and a PLC control part as shown in Fig. 2. The 3D motion command generation part consists of task modules such as an interpolator and acceleration/deceleration controllers. Both linear and circular interpolators are developed by using a reference-word method [6]. The motion controller part consists of a fine interpolator, PID and feedforward controllers to follow a desired path, as well as FIFO(first-in-first-out) communication modules. This module operates at 1ms fixed time interval. Each task is designed so as to operate mutually independent of others. Therefore, user can add new control logics through the MMI modules.

B. Nonreal-time modules

Nonreal-time modules consist of MMI and configuration windows programmed by a graphical user interface based on GTK+ [7]. Users are able to select one of the control laws that have already installed in the Linux kernel and add a new control logic through the MMI module. During the machine cycle, the developed system displays various conditions and status of the system on the monitoring panel such as the current position of each axis, current velocity, and so on. Internal variables of the CNC system, such as interpolation period, controller parameters, and control period, etc. could be changed according to the desired specification of users on the configuration window. Fig. 3 shows the MMI of the

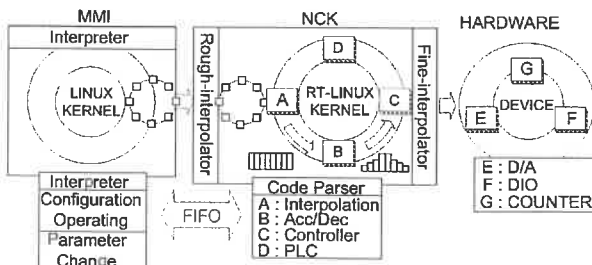


Fig. 2 Structure of numerical control kernel (NCK).

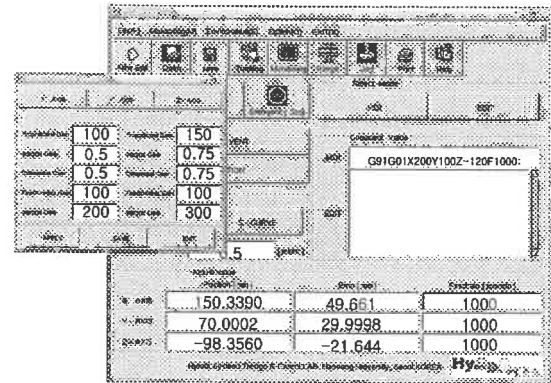


Fig. 3 MMI of the developed system.

developed system.

C. Communication modules

Fig.2 shows the data flow of each task on the Linux-based CNC system. The data created by each task transmits other task with the ring buffers for stabilization. The critical and the non-critical time tasks are communicated through the FIFO channels. For large amount of data, shared memory regions are used for inter-process communication.

D. Hardware components

Basic hardware components used in the developed CNC system are composed of a digital-to-analog converter with 8-differential I/O channels, 8 high-frequency-counter chips, and a 32-bit multi-channel digital I/O board as shown in Fig. 4

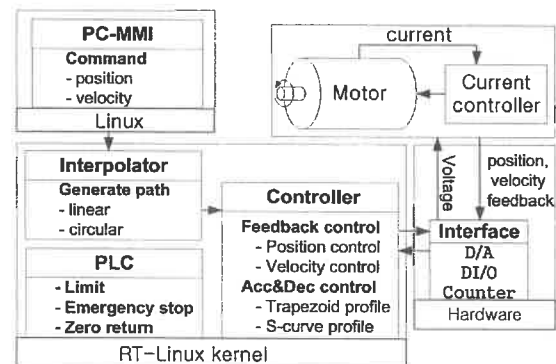


Fig. 4 Configuration of the developed CNC system.

4. Application

In order to verify the performance of the developed system, the Linux-based CNC system has been applied to a precision x-y positioning system and a horizontal arm type coordinate measuring machine. System parameters and controller gains have been

manipulated appropriately by using the Ziegler-Nichols method in each application case [8].

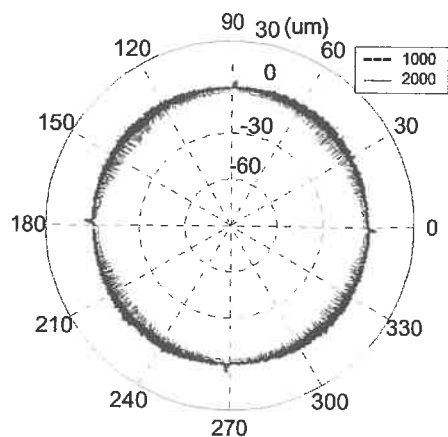


Fig. 5 Result of circular motion error.
(Radius:25mm, Feedrate:1000,2000mm/m)

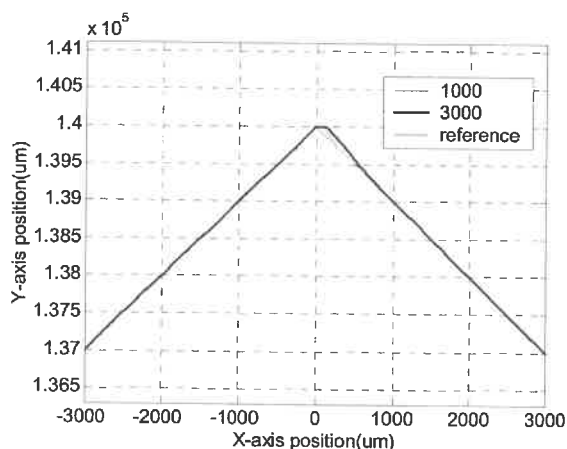


Fig. 6 Result of linear motion error.
(Feedrate:1000,3000mm/m)

A. X-Y table

(1) Linear & Circular interpolator tests

Linear and circular motion errors are measured with respect to various operating conditions in the precision x-y positioning system. As shown in Fig. 5 and 6, we confirmed that the developed CNC system provided a good tracking performance in various kinds of motion.

(2) Cross-coupled control (CCC)

A cross-coupled controller is formed in the real-time module to estimate contour error in two-dimensional motion. Since the CCC module is dynamically loaded into the Linux-based CNC system, user can easily select this controller according to the purpose of control on the MMI. Linear motion

experiments are conducted with various control parameters as shown in Fig 7.

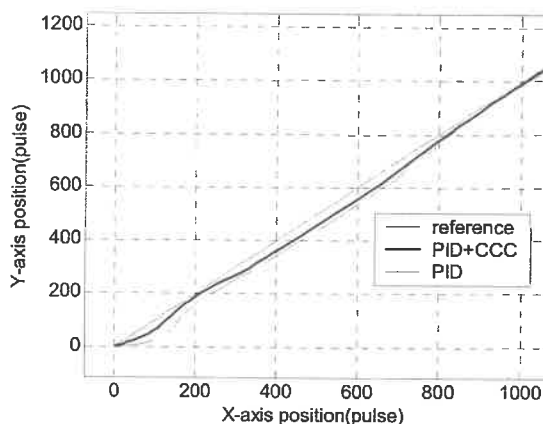


Fig. 7 Result of linear motion with CCC.
(Feedrate:2000mm/m)

B. A horizontal arm type CMM

A horizontal arm type CMM [9] consists of three orthogonal axes is used to verify the measuring and control performance of the Linux-based CNC as shown in Fig. 8.

Owing to the demand for shorter cycle times for measurement tasks, CMMs are required to be used at high speeds. In such high-speed measuring processes, acceleration torques contain high frequency component, which excites the structural dynamics of the driving mechanism and generates undesirable vibration [4]. To reduce the high frequency torque signals, acceleration command profile produced by a linear interpolator should be smooth. The jerk-limited acceleration profile should be included in this system.

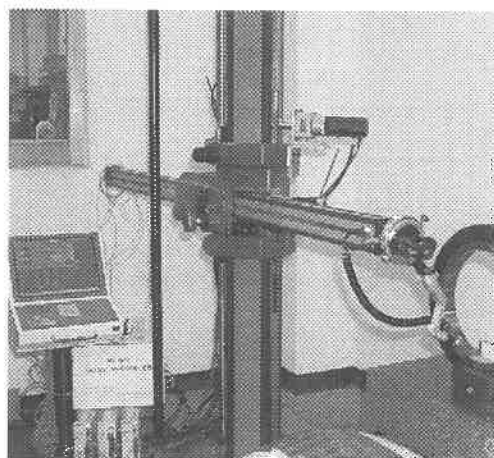


Fig. 8 Horizontal arm type CMM.

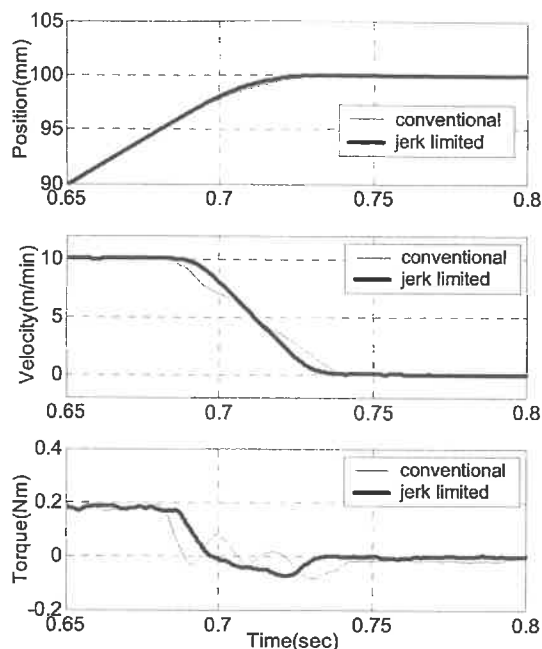


Fig. 9 Comparisons of trajectories between conventional and jerk-limited acceleration Profiles. (Position, Velocity, Torque)

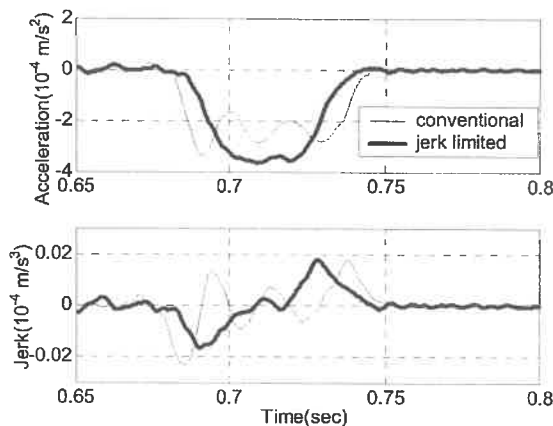


Fig. 10 Comparisons of trajectories between conventional and jerk-limited acceleration profiles. (Acceleration, Jerk)

Fig. 9 and 10 show the linear motion command with jerk-limited acceleration profiles. We can confirm the reduced structural vibration through the acceleration controller.

4. Conclusion

This paper proposes a design methodology for the Linux-based CNC control system with an open architecture. The Linux-based CNC system is designed using commercially available Linux-based

software and several I/O devices such as digital-to-analog converters, counter chips, and a digital I/O board. A GTK+ language is used to develop the MMI modules in the Linux system and various control modules are written in C-language.

The developed Linux-based CNC system is applied to two different motion control systems, which are a precision positioning system and a horizontal arm type coordinate measuring machine.

In order to verify the performance of the developed system, several motion control experiments have been conducted in the application systems with various operating conditions.

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FABRICATION OF AN INCH-WORM TYPE MOBILE MICROROBOT MOVABLE IN DIFFERENT DIAMETER AND LONG PIPES

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1. Introduction

We have many small diameter pipes which are cooling pipes for atomic power stations, boiler pipes for industries, gas and water pipe lines for individual or corporate houses. They must be periodically inspected in order to protect the accident previously. Some of these pipes are long and their diameters are different at the place where pipes change from the main to the branch. The inspection robot for these pipes must move different diameter and long distance.

A step comes where the pipe changes its diameter. The in-pipe robot driven by wheels is very difficult to cross the step [1]. We have used cone-shape friction rings for the driving legs of the in-pipe robot. However, the robot driven by friction rings is also difficult to move in the pipe where diameter changes more than 3 mm [2].

Now, we propose a mobile robot that can surely move in a long pipe whose diameter changes. We are needed to hold the different diameter pipe to the radius direction and to drive the robot to longitudinal direction using the friction force by the holding the pipe, in order to move surely in the different diameter pipe. In order to achieve this motion, we developed a new mechanism which is made of three rubber bellows in series. In the mechanism, two outer rubber bellows are provided eight bulging rubber sheets. These are called as holding elements with braking mechanisms. So, the holding element works as a break of the moving mechanism. The center rubber bellows is called as a driving element, because the rubber bellows drive the moving mechanism by its stretching and

shrinking motion.

When the rubber bellows of the holing element shrinks, eight bulging rubber sheets spread to the radius direction and touch the different diameter pipe. Then the spread bulging rubber sheets surely hold the different diameter pipe. When the rubber bellows of the holing element stretches, spreading to the radius direction of the eight bulging rubber sheets and touching to the different diameter pipe are canceled. The rubber bellows are stretched by air pressure and shrank by vacuum pressure. The stretching and shrinking of the rubber bellows are controlled by a computer and electromagnetic valves.

The inspection robot is required to move long distance in the pipe. However, large time is needed to send air pressure pulse to the air actuators which are laid at the end of long air-feeding tube because of the compressibility of air, if the electromagnetic valve is laid near at the air pressure generators. Robots whose electromagnetic valves are laid near at the air pressure generators could not move long distance[3], [4]. So, we arranged electromagnetic valves, which make air pressure pulse, near the air actuator in the case of the fabricated mobile robot. We confirmed that the new mobile robot can move in the long and different diameter pipes.

2. Structure of fabricated robot and characteristics of elements

An in-pipe mobile robot which is able to move in pipes whose inner diameters are among 64 mm and 90 mm is shown in Fig. 1. The robot consists of two holding elements

and a driving element. The holding elements are driven by bellows which are 33 mm in outer diameter, 23 mm in inner diameter, 55 mm long. The driving element is driven by bellows which is 100 mm long. They are made of nitrile butyl rubber (NBR). Each bellows is an independent vessel. Three electromagnetic valves are arranged near three bellows of the holding elements and the driving element. Consequently, these three electromagnetic valves move with the mobile robot. Two air-feeding tubes which supply air pressure and vacuum pressure and 4 mm in outer diameter, 2 mm in inner diameter are connected to the three electromagnetic valves. The rubber bellows are stretched by the supply of the air pressure and are shrunk by the supply of the vacuum pressure. The stretching and shrinking of the bellows are controlled by three electromagnetic valves and a computer.

A braking mechanism of the holding element is shown in Fig. 2. The braking mechanism consists of eight bulging rubber sheets which are 10 mm wide, 2 mm thick, 80 mm long and made of NBR. The length of the braking mechanism is 80 mm and the outer diameter of the bulging brake is 40 mm in the condition where the pressure in the bellows is +80 kPa. The length of the braking mechanism is 31 mm and the outer diameter of the bulging brake is 100 mm in the condition where the pressure in the bellows is vacuum of -80 kPa. Then the bulging rubber sheets are strongly pressed to the pipe whose diameter is less than 100 mm. The friction force in the pipe are made by the radial force of the bulging rubber sheets.

3. Mobile robot system

An experimental apparatus for measuring the characteristics of the mobile robot is shown in Fig. 3. A computer controls three electromagnetic valves through a valve controller. Three air-feeding tubes are connected from the electromagnetic valves to two holding elements and the driving element of the mobile robot. These electromagnetic valves move with the mobile robot. An entrance ports of the electromagnetic valves through long air-feeding tubes and

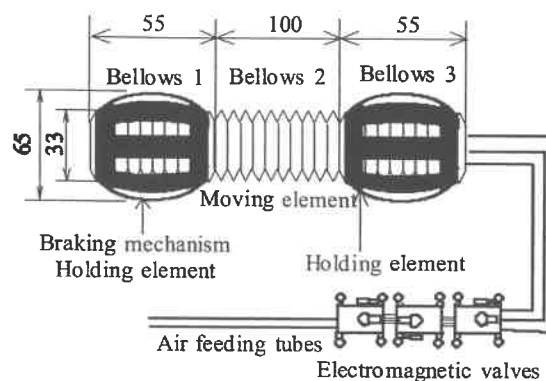


Fig. 1 Structure of the fabricated mobile robot movable in different diameter pipes

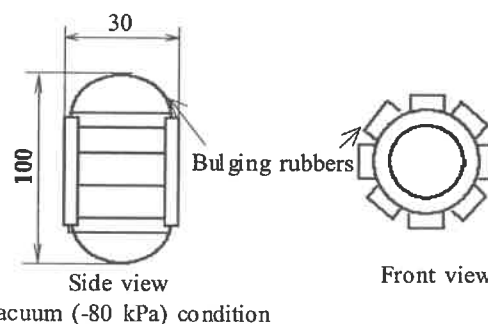


Fig. 2 Structure of the braking mechanism of the holding element

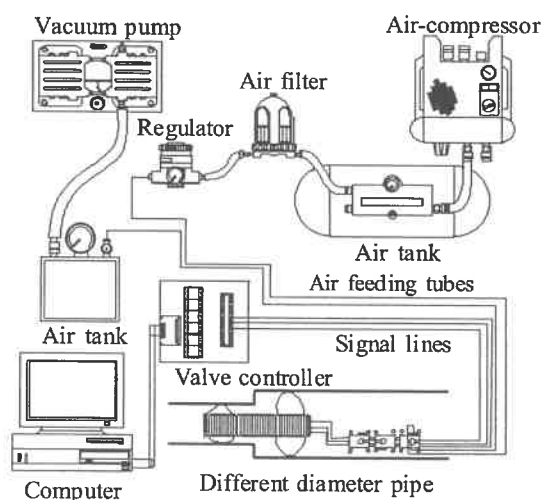


Fig. 3 Experimental apparatus of the mobile robot system

air-compressor is connected to the feeds air pressure to stretch the bellows. A vacuum pump is connected to the exit ports of the electromagnetic valves and feeds vacuum pressure to shrink the bellows.

4. Principle of the moving of the mobile robot

The air and vacuum pressure are fed like as a time-chart shown in Fig. 4. The mass flow rate through the electromagnetic valve is proportional to the absolute pressure of the upstream flow, because the electromagnetic valve is a kind of an orifice. The absolute air pressure at the time of stretching motion is more than two times of the pressure in the bellows in the time of shrinking motion. Then the supplying time of the vacuum pressure in the shrinking motion ($=\tau_2$) must be more than two times of the supplying time of the air pressure in the stretching motion ($=\tau_1$). The sum of τ_1 and τ_2 is called as the cycle time ($=T$).

As shown in Fig. 4, we make a time difference of $\tau_1/2$ in the air supplying time among the front holding element, the central driving element and the rear holding element. The mobile moving motion is shown as Fig. 5, when the air and vacuum pressure are supplied as shown in Fig. 4.

- (1) At initial, all the bellows are shrinking by the vacuum pressure. The braking mechanisms of the front and rear holding element are at the condition of the braking and hold the pipe.
- (2) The air pressure is fed to the front holding element and the braking is free.
- (3) The air pressure is fed to the central driving element and the mobile robot is stretching. Then the front part of the mobile robot can move to the forward direction, because the rear holding element is still at the condition of the braking and holds the pipe.
- (4) The air pressure is fed to the rear holding element and the braking is free. At the same time, the vacuum pressure is fed to the front holding element and it is at the condition of the braking. The vacuum pressure is fed to the central driving element and the mobile robot is

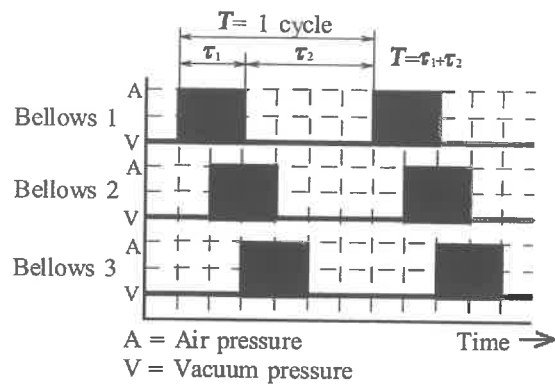


Fig. 4 Time chart of air and vacuum pressure for each bellows

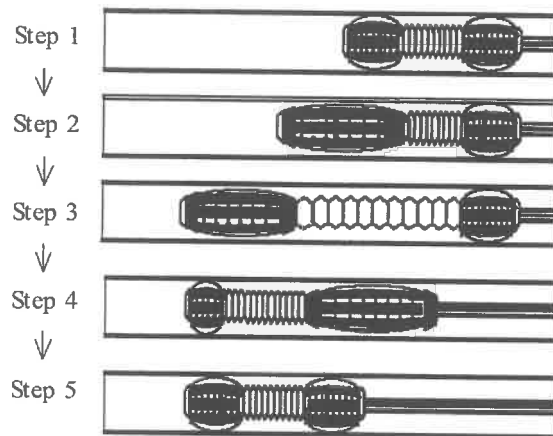


Fig. 5 Principle of the moving of the mobile robot

shrinking. Then the rear part of the mobile robot can move to the forward direction, because the braking of the rear holding element is free.

- (5) All the bellows are shrunk by the vacuum pressure. The braking mechanisms of the front and rear holding element are in the condition of braking and hold the pipe. It is same as the initial condition and one cycle is over. The mobile robot moves the stretching displacement of the driving element. Consequently the speed is shown by (stretching displacement) / (cycle time).

5. Experiments

5.1 Measurement of the traction force

We measured the generated traction force of the microrobot which moves to perpendicular direction. Relationship between the cycle time and the traction force is shown in Fig. 6. The maximum traction force is obtained at the cycle time of 4.2 seconds. The traction force is confirmed to be nearly equal to the friction force of the braking mechanism. Consequently, The mobile microrobot can move distance of 25 m in the case of 90 mm diameter pipe and distance of 45 m in the case of 74 mm diameter pipe.

5.2 Measurement of the speed

We measured the speed of the mobile microrobot supplying their pressure of 0.08 [MPa] and the vacuum pressure of -0.08 [MPa]. The diameters of the pipe are 90 mm, 74 mm and 64 mm. The combinations of the diameter are as follows; 90 mm and 74 mm, 90 mm and 64 mm, 74 mm and 64 mm. Moving is done from larger diameter to smaller diameter pipe. We have steps of 5 mm in minimum and 13 mm in maximum between two pipes. However, the fabricated mobile microrobot could move the steps. Relationship between the cycle time and the speed is shown in Fig. 7. The mobile microrobot was confirmed to move different diameter pipes at the speed of 20 or 27 [mm/s].

6. Conclusions

- (1) We proposed a new mobile microrobot that can surely move in long and different diameter pipes. The microrobot consists of two holding elements and a driving element. The holding element has a braking mechanism which consists of eight bulging rubber sheets.
- (2) The friction force between the holding element and the pipe was measured. Friction force of 9 N is obtained at the large diameter pipe of 90 mm and 20 N is obtained at the small diameter pipe of

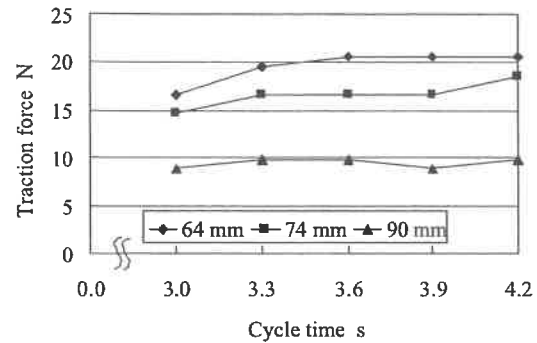


Fig. 6 Relationship between the traction force and cycle time

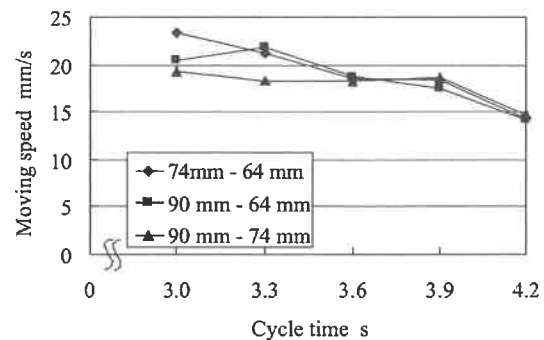


Fig. 7 Relationship between the moving speed and cycle time

64 mm.

- (3) The mobile microrobot was confirmed to move in different diameter pipes whose combinations of the diameter are 90 mm and 74 mm, 90 mm and 64 mm, 74 mm and 64 mm. Its speeds were 20 or 27 [mm/s].

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Fine Motion Performance of L-shaped Seal Mechanism with 3 Degrees of Freedom

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1. INTRODUCTION

Multiple degrees of freedom (DOFs) devices with a wide movable range and a high resolution are required for a fine motion stage in scanning probe microscopes (SPMs). A coarse motion device is generally combined with a fine motion one to build such a positioning device. Inchworm Mechanism [1][2] and Impact Drive Mechanism [3][4] have been developed for a coarse motion mechanism for SPMs. With an increase of DOFs of the devices, the number of controlled actuators is increased, the size of a whole device becomes large and its structure becomes complex.

The authors have proposed Seal Mechanism with 3 DOFs. Though the mechanism has a smaller number of controlled devices, it can move with micrometer order steps in the x -, y - and θ -directions. Though Seal Mechanism has smaller number of controlled actuators than Inchworm mechanism, it performs as well as Inchworm mechanism [5][6]. An L-shaped device of Seal Mechanism has been proposed and its positioning performance in Coarse mode has been investigated [6]. In this paper, positioning performance of Seal Mechanism in the combination of Coarse mode with Fine one is described.

2. PRINCIPLE OF MOVEMENT

2.1 Movement Principle of 3-DOF Device in Coarse Mode

Fig. 1 shows a structure of a 3-DOF device of Seal Mechanism [6]. This device consists of two controlled Friction devices CA and CB, connected by two Extension devices PA and PB with Friction devices CC constant friction. Two Extension devices cross at right angle. Extension devices are alternated by on-off control in Coarse mode. It can move in a wide range. A Friction device CC that generates a constant frictional force works as a passive element. Therefore, it is possible to move with 3 DOFs by using two Friction devices and two Extension devices as the controlled elements. The following relation must be satisfied:

$$F_{xoff} < F_c < F_{xon} \quad (1)$$

where F_c is a constant frictional force at Friction device CC, and F_{xon} and F_{xoff} are frictional forces at Friction device CA or CB in the cases of adhering and releasing, respectively. Fig. 2 shows the movement principle of a 3-DOF device. The device can move in the $+x$ -direction as shown in Fig. 2 (a). Figs. 2 (b) and (c) illustrate the movement principle in the y - and θ -directions. Either PA or PB is used to rotate in the θ -direction.

2.2 Movement Principle of 3-DOF Device in Fine Mode

The step like movement can be interpolated by changing the length of Extension device continuously, which is similar to Walking Drive Mechanism [7]. Friction device CC can move when this device adhered onto a base with CA and CB, and two Extension devices expand and contract continuously. It can draw various loci with the input signal. For example, a locus will become a circle when a sine wave is applied to PA, a cosine one is applied to PB simultaneously. 2 DOFs, the x and y -directions are controllable in this mode.

3. STRUCTURE OF DEVICE EMPLOYING SEAL MECHANISM

Fig. 3 shows an appearance of a 3-DOF device. Stacked piezoelectric actuators and electromagnets were used for Extension devices and Friction device, respectively. The whole device measures $63 \times 63 \times 16$ mm and weighs 34 g. The piezoelectric actuator with dimensions of $5 \times 5 \times 20$ mm expands $11.6 \mu\text{m}$ at the applied voltage of 100 V.

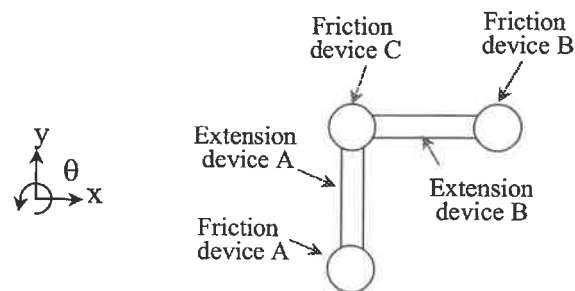


Fig.1 Structure of Seal Mechanism with 3 DOFs

The waveforms were given through drive amplifier generated with a personal computer. The signal from an additional capacitor to pick up noise was subtracted from the signal from the charge amplifier to reduce the noise from power lines in the induced charge feed back control [8][9]. The phase of the signal from the pick up capacitor was adjusted by using a filter to each piezoelectric actuator.

Displacements were measured with eddy current displacement sensors with a measurement range of 1 mm and a resolution of 0.4 μm , and capacitance displacement sensors with a measurement range of $\pm 25 \mu\text{m}$ and a resolution of 8 nm. A rectangular wave was applied to Electromagnets CA and CB. An applied voltage to Electromagnet CC was determined to 1 V experimentally.

4. EXPERIMENTS

4.1 Displacement control by feedback of induced charge

The elliptically moving device consists of two piezoelectric actuators crossing at right angle. When the voltage of a circular locus was applied to it, its tip moves elliptically because of an isotropic stiffness and the hysteresis of piezoelectric actuators [10][11]. This device had some interference among the axes in Fine mode. Therefore, displacement control by the feedback of the induced charge was

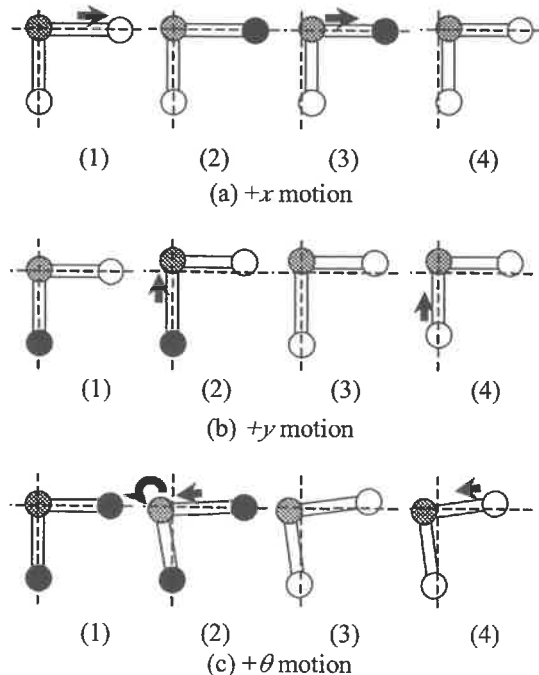


Fig.2 Principle of movement of Seal Mechanism

tried. Equation (2) predicts the displacement of this device from the extension of the piezoelectric actuators.

$$x = AC = \begin{pmatrix} 3.71 & 0.39 \\ 0.24 & -2.71 \end{pmatrix} \begin{pmatrix} C_{PA} \\ C_{PB} \end{pmatrix} \quad (2)$$

where $x = [x \ y]^T$ is a position vector with 2 DOFs, A is a coefficient matrix obtained from experimental results, and C indicates the induced charges vector with 2 DOFs. The generalized inverse matrix of C is used for the displacement control.

Fig.4 shows an example of positioning when the reference positions were set to $x = -1 \mu\text{m}$ and $y = 1 \mu\text{m}$ at the same time. In 10-time positioning, the maximum errors in x and y -directions were 0.09 and 0.11 μm , respectively. Noise, impulsive force and stick-slip caused this error.

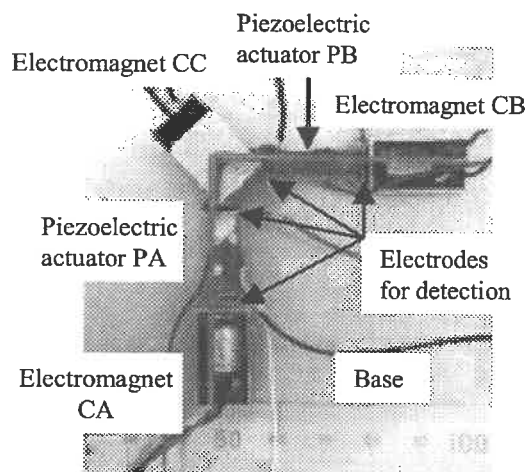


Fig.3 Appearance of Seal Mechanism with 3 DOFs

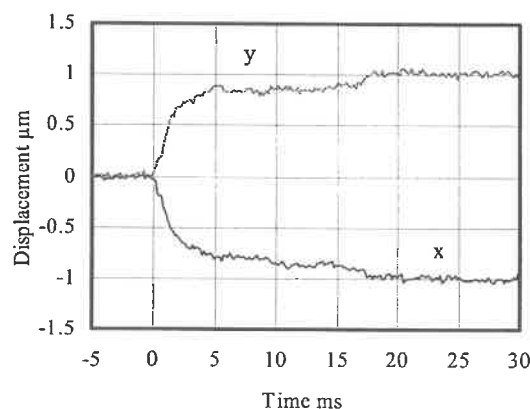


Fig.4 Positioning by induced charge feedback control

4.2 Coarse and Fine positioning by displacement feedback control

Positioning by the feedback of the displacement was carried out by the combination of Coarse and Fine modes in absolute coordinate system. The mode is switched from Coarse to Fine when the device reaches the movable range of Fine mode in the x - and y -direction and the tolerance in the θ -direction. The displacements was measured with eddy current displacement sensors. The reference positions was independently set to $x = 100 \mu\text{m}$ or $y = 100 \mu\text{m}$ and the others was set to zero. Fig. 5 shows an example of positioning in the y -direction. Fig. (a) shows the whole positioning process and Fig. (b) shows expanded at switching from Coarse mode to Fine one. Table 1 shows results of 10-time positioning in each direction. Positioning accuracies in the x - and y -directions were smaller than the resolution of the displacement sensor. The error in the θ -direction was less than the tolerance in the θ -direction in Coarse mode.

4.3 Control of circular motion

The circular motions of 10 revolutions were carried out after the applied voltage was gradually raised to reduce the impulsive force at its beginning. Fig. 6 shows results at a frequency of 3.3 Hz. Table 2 shows results of tracking by the open loop, the displacement feedback and the induced charge feedback control. Because of the influence of the noise, the induced charge feedback could not perform with the same accuracy as the displacement feedback control. Therefore, the treatment of the noise is required in order to improve the motion accuracy by the induced charge feedback.

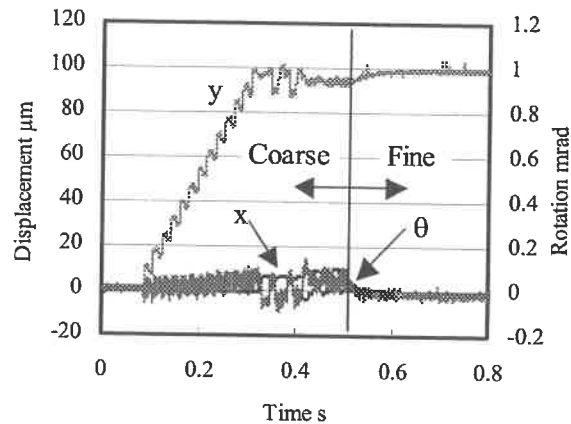
5. CONCLUSIONS

In this paper, the performance of Seal Mechanism in Fine mode was investigated experimentally and it positioned by the combination of Coarse mode and Fine one. The conclusions can be drawn as follows.

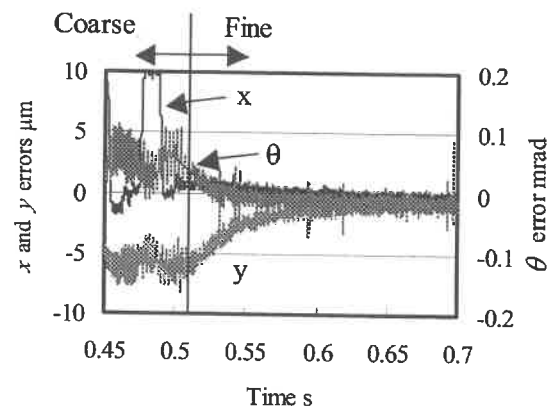
- (1) The relative positioning within an extendable range of the piezoelectric actuators was possible by the induced charge feedback.
- (2) The absolute positioning that combined the coarse mode and fine mode was possible by the displacement feedback.
- (3) Only 2 DOFs, the x - and y -directions were controllable in Fine mode. However the final error of the θ -direction is less than the tolerance in the θ -direction in Coarse mode.

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(a) Whole process



(b) Switching from coarse mode to fine

Fig.5 Positioning by displacement feedback control

Table 1 Results of positioning

Reference position		$x=100 \mu\text{m}$		$y=100 \mu\text{m}$	
		μ	σ	μ	σ
Error	$x \mu\text{m}$	0.0	0.2	-0.1	0.1
	$y \mu\text{m}$	0.0	0.3	0.0	0.3
	$\theta \text{ mrad}$	0.014	0.024	0.021	0.019
Time ms	Coarse	581	161	285	57
	Fine	292	70	237	65
	Total	872	156	523	103

μ : average,

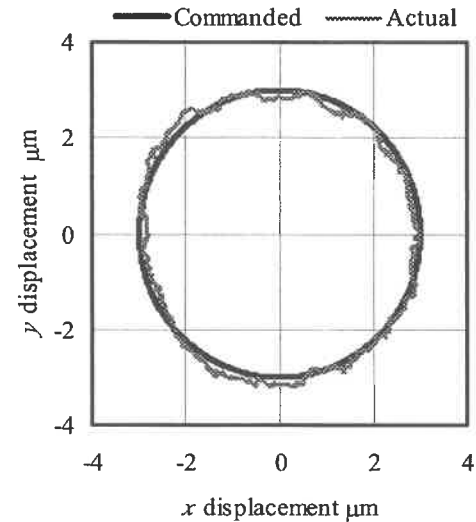
σ : standard deviation

Ministry of Education, Sports, Culture, Science and Technology, Japan and Grant-in-Aid for Scientific Research (C) (2) (13650289) from Japan Society for the Promotion of Science.

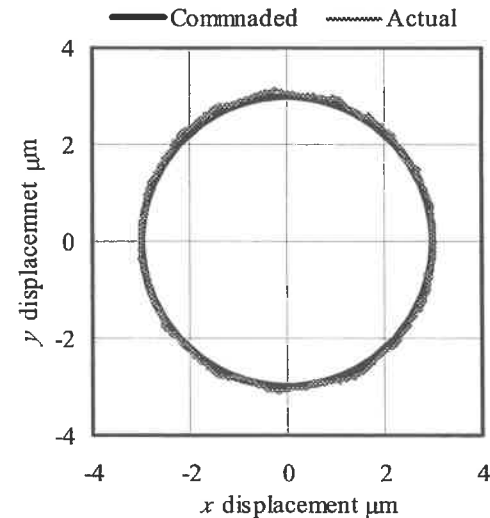
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(a) Induced charge feedback control



(b) Displacement feedback control

Fig.6 Circular motions

Table 2 Accuracy of circular motion

Feedback	Radius μm	Roundness μm
None (open)	2.90	0.87
Displacement	3.05	0.26
Induced charge	3.06	0.51

Design and Development of Micro Hopping Robots for Rescue Operation

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Abstract In this report, the micro hopping robots which have a simple locomotion mechanism and sensory elements have been developed to dispatch into the craves of rubble to detect the alive human. This small robot consists of micro eccentric motor for hopping and thrusting with the help of resonating vibration, and micro sensory element, button battery as well as electronics for detecting the signal source. Moreover, this tiny robot has the ability of the self-righting to keep moving to the target even if it drops from the step. In the primary experiment, many small robots with optical micro sensor have been designed and developed to demonstrate that they can move toward to the light source under the dark environment and the performances such as reliability and durability are discussed.

1. Introduction

As part of rescue operation for the victims under the rubbles, the well-trained dogs are employed to detect the location of victims based on the keen smell sense of the dog. The number of such special dogs, however, is limited and they cannot keep their concentration for hours. And also it is a high risk and dangerous that they encounter the second disaster, such as re-collapse of buildings and the fire. In fact, there are some cases that met with an accident during working. Therefore, need to development of the new supporting device for rescuing the victims under the rubbles, instead of the trained dogs. Recently there are many types of robots have been researched and developed in the universities, institutes and companies, and now on going development^{(1), (2), (3)}

Therefore it can be suggested that the employment of the multiple micro robots for rapid finding out the victims. They can dive into the aperture such as narrow and dangerous space in which the human, the trained dogs and the large size robots can never get. And small disturbance can suppress the second disaster that cause by moving rubbles. Also the mass production of low-cost micro robots can provide the great benefit to the large-scale employment for disaster with widespread investigative area. The aim of this project is to design and to development of micro robots system to find out the victims rapidly under the rubbles as shown in Fig.1. There are many small robots can move into the complex and search the living victims autonomously, then provide the appropriate signal to identify the location.

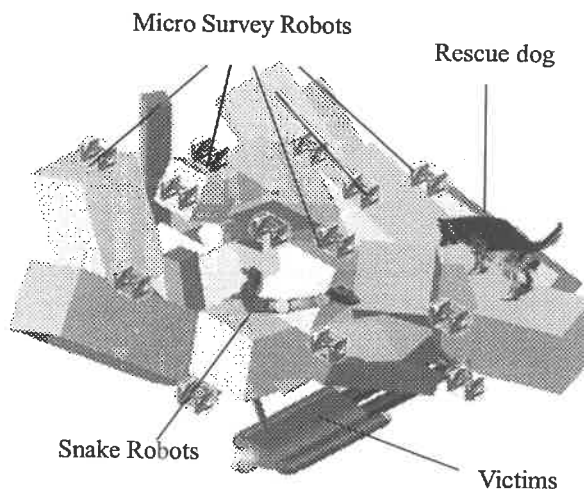


Fig.1 Typical rescue operations at disaster location

2. Design of Micro Hopping Locomotion Mechanism

2.1 Design

To achieve our goal as stated above, it is important that the small robot with the performance to get into the rubbles and to search the victims should be designed and developed at first. The properties at least required to the small robot are summarized below;

- (1) Small size within 40mm x 40mm x 40 mm
- (2) Low cost less than 1,000 Japanese yen

- (3) Disposable use
- (4) Self-contained or self-generation of electricity
- (5) Sensory device and control & guidance logic on-board for simple navigation

Based on these properties, the mechanism for locomotion should be considered carefully and also discussed with the viewpoint of performance, material and device costs.

2.2 Locomotion principle and actuators

From the limitation of the space for loading the battery on the small robot, the mobility even on the rough terrain as well as the highest performance of power consumption are strongly required. So the use of simple resonating mechanism excited by the vibration resource should be one of the solutions to provide the maximum of movement efficiency. As the first stage of the project, the simple locomotion mechanisms with the simple resonator of elastic cantilever are considered as shown in Fig.2. The flat micro vibration motors with the eccentric rotor is set at the elastic simple supported beam. Then this layout can allow the mechanism to move and hop cyclically. As the elastic simple supported beam is attached on the body with the tilt of appropriate angle, the trusting forced which is generated is alternated between forward and backward. The asymmetric force in them can allow the locomotion mechanism to move in one direction.

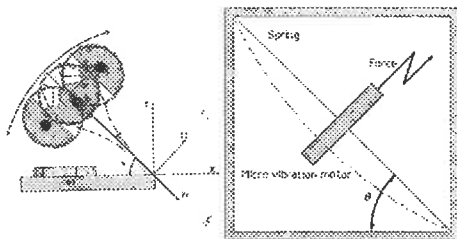


Fig.2 Isometric view of proposed hopping mechanisms; Simple type with one motor in the left and the closed body type with two motors in the right.

2.3 Locomotion dynamics modeling and simulation

In order to get the geometric dimension of the frame with the best performance, the dynamic model of the locomotion mechanism was considered and numerically simulated to check the mobility. For discussing the dynamic equations, the typical results of the active forces, acceleration and velocity are plotted in Fig.3. It is find that the friction that influence from the slipping or jumping on the surface becomes small as the centrifugal force, while the vertical force is increased by the part of centrifugal force caused from micro vibration motor.

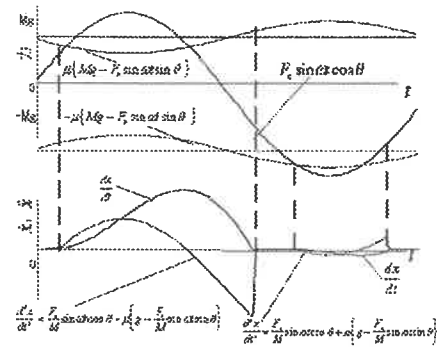


Fig.3 Analysis of locomotion dynamics

This means the friction that influence from the slipping or jumping on the surface becomes large as the centrifugal force. When the movement cycle continues with the alternative of forward and backward, the friction force of forward is smaller than that of backward, and the mechanism can go forwarding.

3. Development of Micro Hopping Mechanism

3.1 Self-righting property

Considering of the requirement that the actual application in the field, the micro robot needs to have the ability of self-righting on anywhere. So the main frame with two motors and two batteries has the square frame at each end and the guard ring made of thin elastic wires for preventing from getting stuck as illustrated in Fig.4. This structure can provide self-righting with the help of wire frame and stable contact with the surface at any orientation when it falls from the step.

Guard (thin elastic wire)

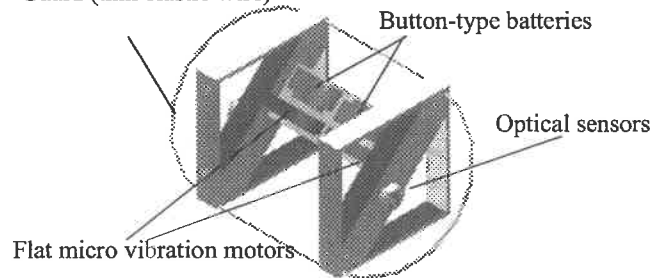


Fig.4 Illustration of micro hopping robot on-boarded electronics devices with wire ring for self-righting.

3.2 Sensor

In this part, it is explained that the sensory devices, their control electronics and guidance logic for simple navigation. As stated in the introduction, the micro robots need the special properties to investigate the

victims under the rubbles in rescue operation. As the first primitive step of development, the simple optical based guidance system is incorporated so that the micro robots can approach to the light source automatically. In the experiment, the CdS type optical photocells are chosen and the electrical circuit as shown in Fig. 5. This simple electronics without CPU can switch the electric current to the motor according to the detected amount of light so that the small robot can move toward the optical source. The ability of turning mechanism is shown in Fig.6.

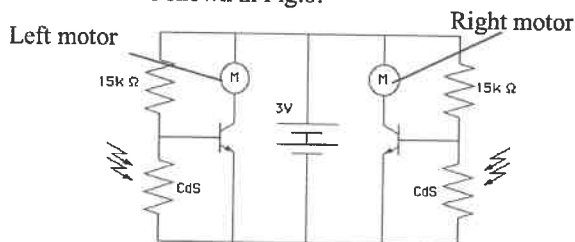


Fig.5 Electrical circuit for switching motor current based on sensor signal

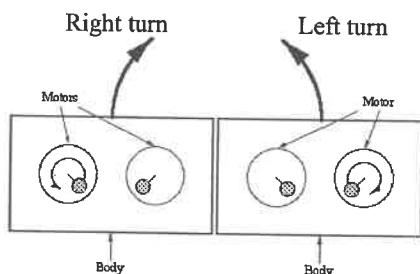


Fig.6 Ability of turning left or right

4. Experimental Results

4.1. Moving motion speed

The micro hopping robot with two motors is shown in Fig.7. This robot is incorporated with a leaf spring for the improvement of robot's speed performance. As the result of measurement of both type robots (with spring mount and rigid mount), the motion speed with respect to motor driving voltage of the small motor is plotted in Fig.8. When the voltage rate is ranged from 1.2[V] to 3.3[V], the maximum motion speed of spring mount robot was achieved 60[mm/s] at 3.3[V] and rigid mount robot was achieved 48[mm/s] at 1.8[V]. The tilting angle was fixed 45 degree.

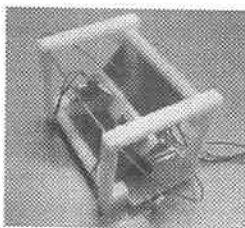


Fig.7 Micro hopping robot with two motors on leaf spring

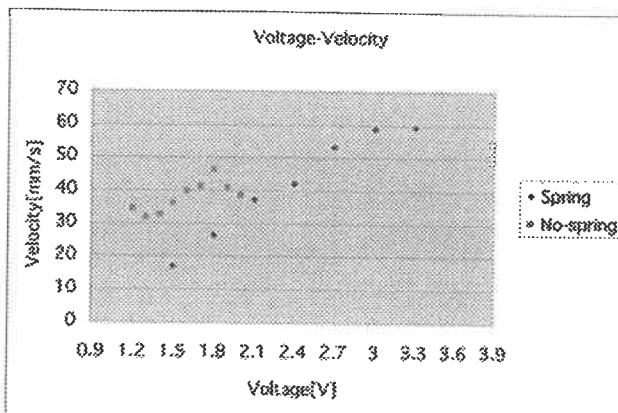


Fig.8 Motion speed of two motors hopping mechanism

4.3 Durability

The durability of the small robot with the button battery was also interested and checked in the experiment as shown in Fig.9. When the battery of LR44 (1.5V, 100mAH) is used to drive the motor, it can supply the electricity over 340 minutes, while CR1616 (3.0V, 60mAH) can do over 100 minutes. Depending upon the situation, the durability over approximately 2 hours should be enough to operate such the small robot system for the special task such as rescue.

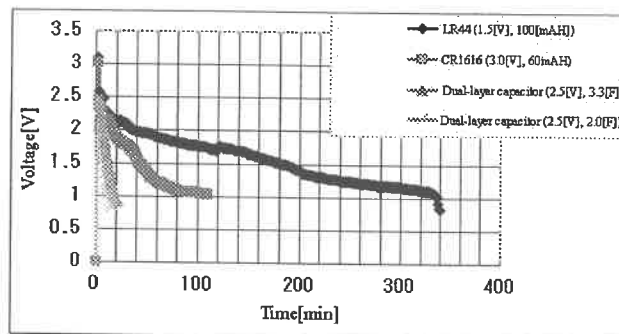


Fig.8 Durability of the battery

4.4 Automatic navigation to signal source

In Fig.9 (a) (b), they were gauged by camera based tracking measurement instrument and the single small robot could move toward the optical source automatically from any starting orientation. Many tracked lines showed that the repeatability was acceptable although it may lose the signal source due to the noise or the reflection sometimes.

In Fig.10, the sequential photo that three small robots could move toward to the optical source in the dark environment after falling from the chair of 50cm in height. It seemed that one of them may lose the signal source, although much more small robots such the number of 100 or 1000 can cover the wider range

of the field and the random trooping by many small robots is rather effective for detecting the victims.

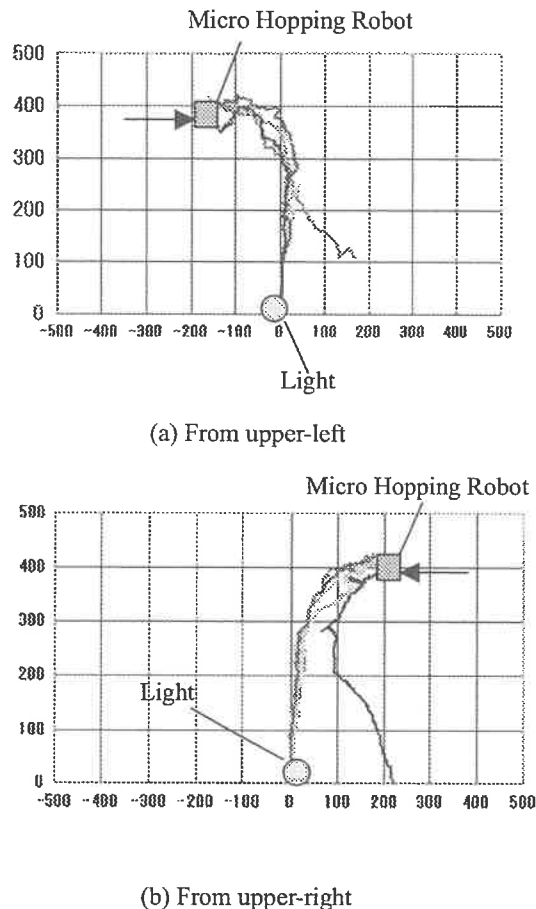


Fig.9 Automatic navigation to light source

5. Conclusion

In this paper, it was proposed that many small hopping robots system could be employed for detecting the victims under the rubbles as the rescue operation. To achieve this task, the micro hopping locomotion mechanism was considered at first. This locomotion mechanism was composed of the simple structure that was excited by the micro eccentric motor. The asymmetrical force could trust and lift the body and then it could move with hopping. Also the simple prototype small robot incorporated with the optical sensor, electronics and battery was developed. After measuring the basic performance of the small hopping robot, it was successful in navigating these small robots to approach to the static signal source of light even after falling from the top of the table.

As an acknowledgment, a part of this project is financially supported by AOARD (Contract No. 02-4008).

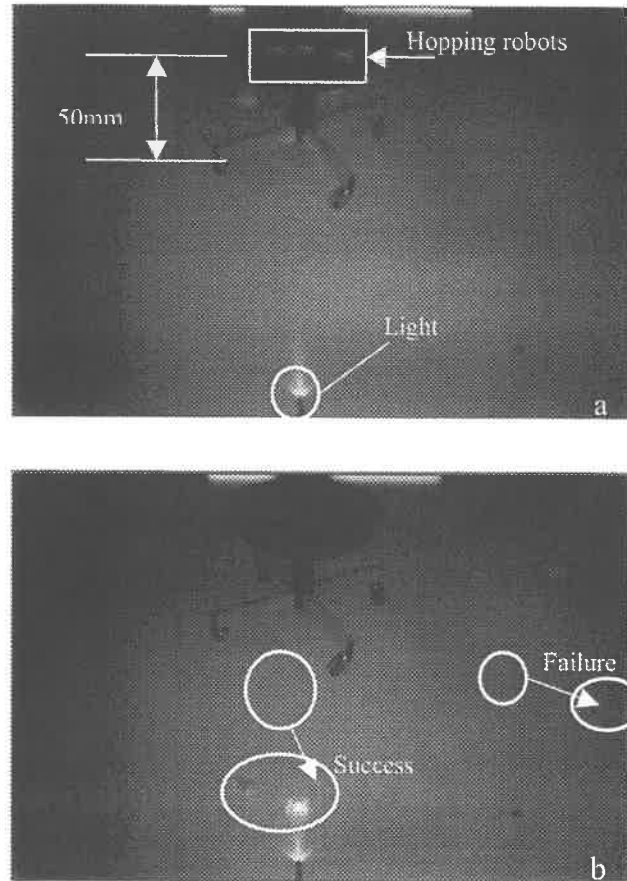


Fig.10 Automatic navigation of three small robots to approach the light source after falling from the step of 50cm in height

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A simple hybrid positioning stage with <1nm resolution

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1. Background

A variety of ultra-precision positioning devices are nowadays effectively in use, complying servomotors, no matter rotary or linear. On the other hand, step motors, or pulse motors, have position holding characteristics in nature and capable to control position without feedback devices. So-called "micro-step drive" is an effective method to extend the positioning resolution of step motors, which controls phase current stepwisely and interpolates the basic step angle. Step motors and drives are relatively compact and cheap. Even though, step motors are still lacking resolution and accuracy, as well as smoothness in motion, for nanometer-order motion control.

One approach is to take those advantages and to compliment those shortages would be utilization of a piezoelectric actuator. Namely, the step motor acts as a coarse actuator driving the stage in a long range, while the piezoelectric actuator as a fine actuator compensates the motion error of the coarse mechanism and interpolates the resolution of the step motor. Simple and cheap mechanical stages with nanometer-order resolution have potential to be applied in many area.

Such hybrid stage systems are already commercialized¹⁾. In some applications, continuous motion control is needed as well as fine positioning. The purpose of this study is to apply a continuous motion control system to the stage mechanism and to evaluate the characteristics with nanometer resolution.

2. Mechanical stage

The hybrid stage used is shown in Fig. 1, and its structure is schematically shown in Fig. 2. The stage is driven by a step motor via a ball screw of 1mm lead, guided by a pair of crossed roller guides. A PZT actuator is mounted at the other end of the screw, which pushes back the screw, compressing the coupler, and gives slight

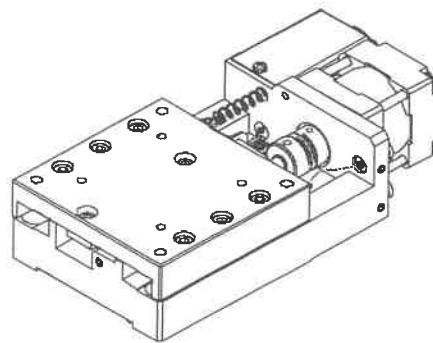


Fig.1 A hybrid mechanical stage

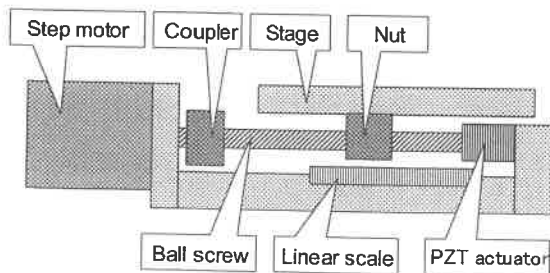


Fig. 2 Schematic diagram of the hybrid stage

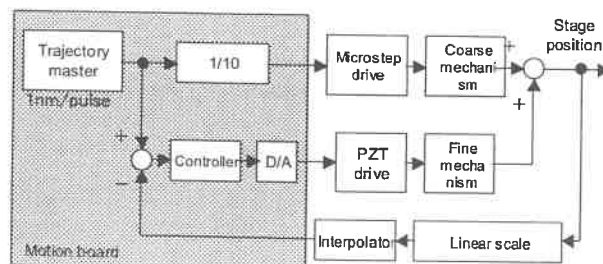


Fig. 3 Schematic diagram of the control system

displacement to the stage. A novel tiny optical linear scale²⁾ is embedded in the stage, which measures the stage displacement with <1nm resolution. The mechanism sizes 60 mm wide and 129 mm long.

3. Control system

The main control scheme shown in Fig. 3 is realized by a DSP-based motion controller, including trajectory generation, pulse distribution, PID controller and analog output for PZT driver. The resolution of command and feedback from the linear scale is set to 1nm. The 5-phase step motor has 0.72 degree of basic stop size. A micro-step driver interpolates the resolution by 200 times, which makes 10nm of coarse positioning resolution. The coarse system is open loop and the fine system is closed-loop. The both displacements of the fine- and coarse system are added mechanically.

Ideally, when the loop gain of the fine system is infinite, the stage displacement would coincide to the command. Actually, the loop gain has a limit due to a compliance at the coupler and the ball nut. This control scheme is a fixed one without any mode change, programmed on a DSP board.

4. Results

4.1 Response of the sole coarse system

Fig. 4 and Fig. 5 show responses of the bare coarse system, with the fine system disabled, for 10 nm pulses issued at every second. The following features are recognized:

- 1) Overshoot of 7 to 15 nm arises at the step jerk, and it takes about 100 ms to settle.
- 2) Step size is not constant, changing sinusoidally with period of 500 nm. The amplitude of the deviation from ideal step size reaches to +/- 60 nm. A period contains a stationary region equivalently of 70nm where no displacement is generated (marked by "S"). The period of 500 nm corresponds to a quarter of basic step size of the motor, 2 μ m.
- 3) Another stepwise deviation of about 20 to 50 nm occurs by a period of 2 μ m (marked by arrows). This

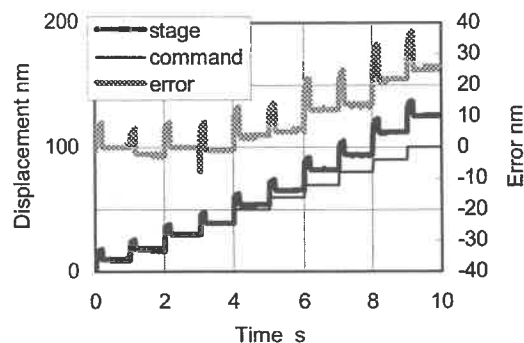
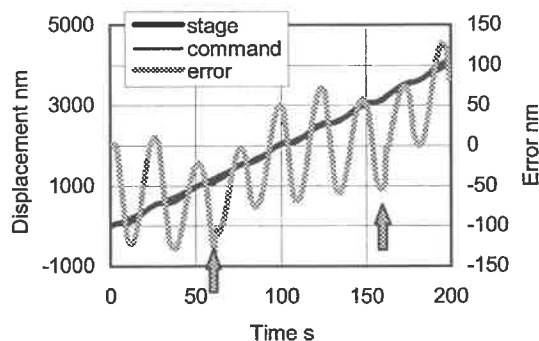
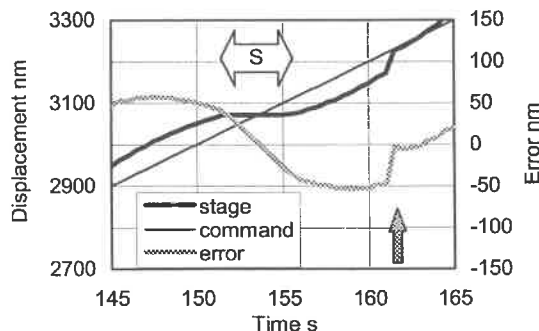


Fig. 4 Response of the sole coarse system

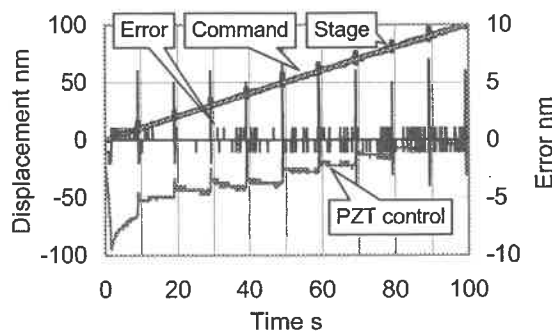


(a) Overview

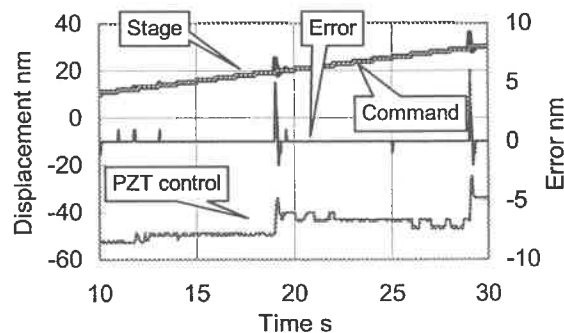


(b) Magnified

Fig. 5 Periodic deviation of coarse system



(a) Overview



(b) Magnified

Fig. 6 Response of hybrid system (1nm step, 1nm/s)

periodic deviation is out of phase with the deviation of 500 nm period.

4.2 Responses of the hybrid system

(1) At low and medium speed

Responses of the fine-coarse system were measured, giving various command step size and step rate, measuring stage displacement, following error and PZT control signal.

Fig. 6 shows the response for a 1nm step train command given at every second. At every step edge the fine system is activated to interpolate the resolution. The deviation is less than 1 nm quasi-statically, but at the every step edge, spike following error arises as overshoot and following undershoot. These are due to the limited bandwidth of the fine system.

Fig. 7 (a) is for a similar condition, while pulse rate is ten times higher. The settling time is measured again as 100 ms, which corresponds to the transient error of the coarse system. This following error is here to be called "FE-A". Another following error is seen, which is due to the stationary region of the coarse system response described in the previous section, and is not compensated perfectly. This following error is here called "FE-B".

Fig. 7 (b) is in such case that command step size is 10 nm. The following observation is made:

- 1) Undershoot of the following error-A is greater than the case at low speed.
- 2) Amplitude of FE-A changes sinusoidally with period of 500 nm. FE-B occurs where FE-A is minimized.
- 3) PZT control as command to fine system also changes periodically, but not sinusoidally.
- 4) Another pulsive following error arises when displacement jump occurs at every 2 μm . This is here "following error-C (FE-C)".

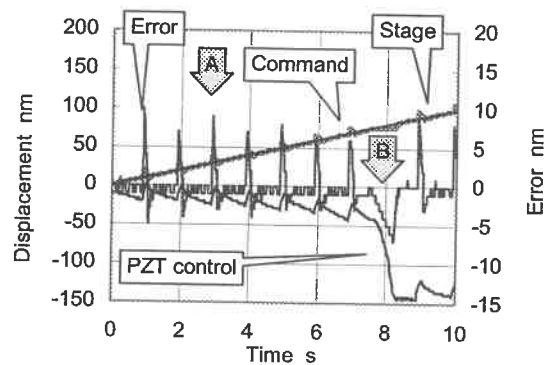
Observing responses under other conditions, velocity and step size were found affecting the degree of the following errors: In Fig. 7 (c) peak values of FE-B and FE-C are increased, as well as width of FE-B.

Peak change of each following errors are plotted as function of velocity and step size, in Fig. 8, where "A+" and "A-" denote positive and negative maximum peak of FE-A respectively, and "B" and "C" stand for maxima of following error B and C respectively. The results is summarized as follows:

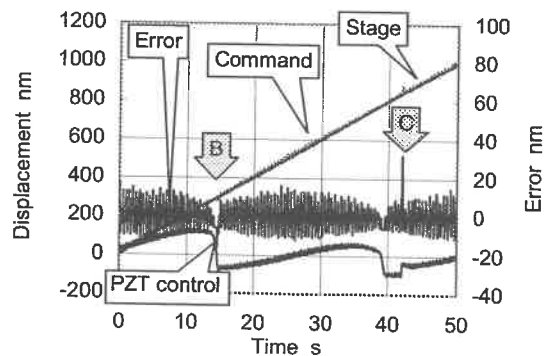
- 1) FE-A+ nor FE-A- does not change very much.
- 2) FE-B bursts when step size exceeds 10 nm.
- 3) FE-C bursts as step size or velocity increases.

(2) At high speed

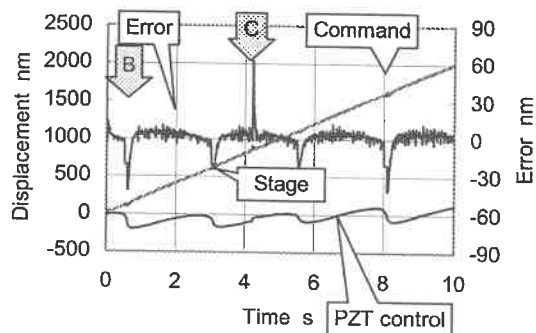
As command velocity is raised into high velocity region, following errors-B and C increase. Peak value and width of the FE-B burst to be dominant factors of following errors. Also, the periodic deviation of step size of the coarse system with 500 nm period appears as waviness in the following errors (Fig. 7(c)). This phenomena is due to the fine system does not compensate the error of the coarse system effectively in higher speed region.



(a) 1nm step, 10nm/s



(b) 10nm step, 20nm/s



(c) 20nmstep, 200nm/s

Fig. 7 Response of hybrid system

(3) Effects of integral loop gain

The following characteristics of the fine system is influenced by its servo loop gain. Proportional gain Kp is set as stable limit. Integral gain Ki has some margin to be increased.

Fig. 9 is comparing the system following error with regular $Ki=50$ and when increased up to 400. As Ki is increased, the following error-B and periodical undulation of following errors were suppressed, while step size as 50 nm, velocity as 2.5 $\mu\text{m/s}$. But when the command velocity was decreased back to 100 nm/s keeping $Ki=400$, another type of following error like oscillation was observed, which differs from transient error seen in previous measurements. Amplitude of the following error is not greater than ever, but the oscillation remains in low velocity region and even under stationary condition. That indicates that higher integral gain is effective only in higher velocity region.

5. Conclusion

Dynamic servo tracking characteristics has been evaluated of a simple hybrid mechanical positioning stage system comprising a pulse motor and a PZT actuator. Fixed control scheme and parameters were applied with 1 nm system resolution. Results are summarized as follows:

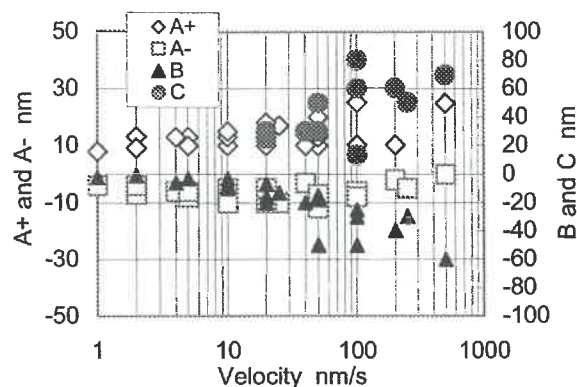
- 1) Quasi-static positioning error is less than 1 nm.
- 2) In dynamic motion control, some transient following errors reside, and they fail smoothness of the motion.
- 3) The limited resolution and variation in step size are effectively compensated by the fine motion system.
- 4) Stepwise and periodic large deviation of the coarse system is not compensated. This imperfection increases as velocity or step size increases.
- 5) It is effective to increase the integral closed loop gain of the fine system for suppressing following error in higher velocity region, while high integral gain introduces instability in low speed region.

6. Acknowledgement

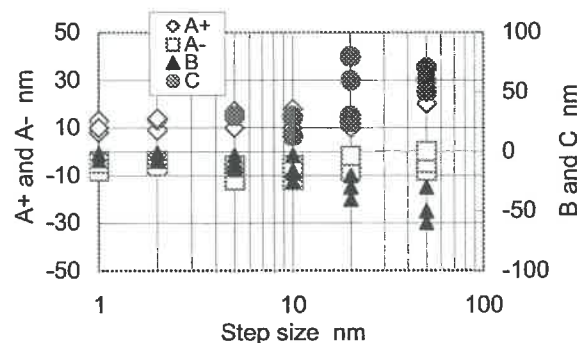
This study has been performed as a part of a Japanese national program, "Regional Consortium Research Development in FY 2001" supported by Ministry of Economy, Trade and Industry, Japan.

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- 2) Japanese patent, No. 3198019

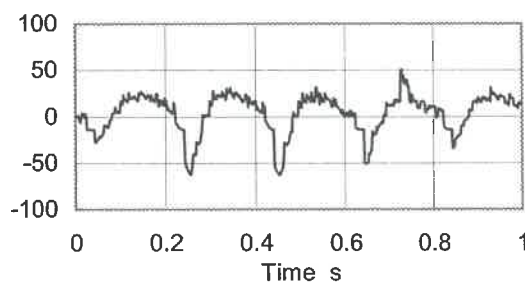


(a) By command velocity

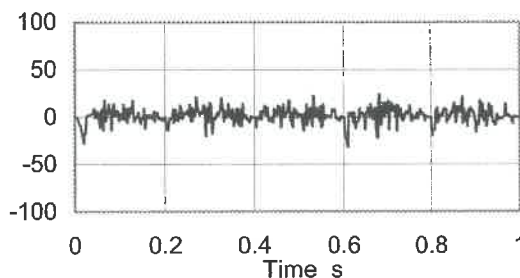


(b) By step size

Fig. 8 Deviation change



(a) $V=2500\text{nm/s}$, $Ki=50$



(b) $V=2500\text{nm/s}$, $Ki=400$

Fig. 9 Effect of integral gain Ki (50nm step)

FABRICATION OF AN ARTIFICIAL EARTHWORM TYPE MOBILE INSPECTION ROBOT MOVABLE IN 100 M LONG PIPES

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1. Introduction

We had an accident that radioactive cooling-water flowed out from the first cooling-water pipe of an atomic power plant, a few years ago. If we can insert an inspection robot and inspect the pipe, we are avoidable from the difficult removal of the heat insulating materials and radioactivity of the flowed water. The cooling-water pipe is long and 76 mm in inner diameter and has vertical parts. Robots driven by electric motor are difficult to move in the vertical pipes, because the ratio between the power and the weight of the electric motor is not large [1]. Authors have researched a new inspection robot which is able to move in the long pipe which has vertical parts and driven by pneumatic actuators, because the pneumatic actuators are advantageous concerning the ratio between the power and the weight. However, they could not move the long distance [2],[3],[4]. This is due to the compressibility of the air. We fabricated an artificial earthworm type robot in which we use three rubber bellows as pneumatic actuators. Twenty rubber friction rings are used to get friction force between bellows and the pipe. Three electromagnetic valves which control pneumatic pressure to the three bellows are attached direct behind of the robot in order to evade the flowing-lag of the pneumatic pressure. The electromagnetic valves and air-feeding tubes are carried on trolleys in order to decrease friction force

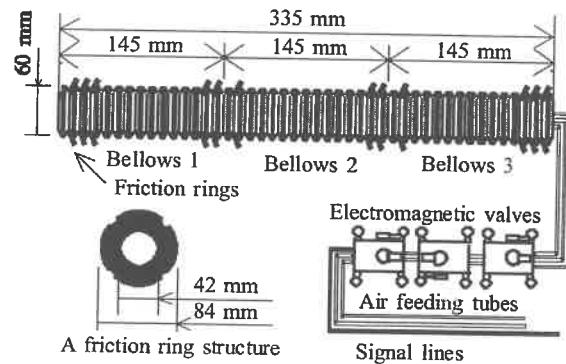


Fig. 1 Structure of the fabricated mobile inspection robot

between the pipe and them. The robot was confirmed to move in the pipe which is 100 m long.

2. Structure of the fabricated microrobot

An in-pipe mobile robot which is able to move in pipes whose inner diameter is 3 inch is shown in Fig. 1. An actual diameter of the pipe is 79 mm. The robot is constructed by three bellows for the pneumatic actuators, friction rings and three electromagnetic valves. Outer diameter, inner diameter and natural length of the bellows are 60 mm, 43 mm and 145 mm, respectively. The bellows are made of Nitrile Butyl Rubber. Total 20 friction rings are attached at the end of the three bellows as moving legs of robot. The friction rings are 84 mm in outer diameter, 42 mm in inner diameter and 1.5 mm thick. When the friction ring is attached at the bellows, the inner diameter is enlarged,

because the inner diameter of the friction ring is 3 mm smaller than the inner diameter of the bellows. Then, friction ring comes to the shape of a cone. Three electromagnetic valves which feed pneumatic and vacuum pressure are attached at direct behind of the actuators. Consequently, the electromagnetic valves move with the actuators. An air-feeding tube made of polyurethane (inner diameter is 4 mm) is attached to the electromagnetic valves in order to feed pneumatic pressure. Another air-feeding tube (inner diameters is 4 mm) are attached to the electromagnetic valves in order to feed vacuum pressure. A power line for the electromagnetic valves is bundled with air-feeding tubes. They are 120 m long.

The air-feeding tubes are carried on conveying mechanisms in order to decrease friction force between the pipe and them. The air-feeding tubes conveying mechanism consists of a chassis made of vinyl chloride pipe, 8 legs made of aluminum and 8 wheels made of plastics. The wheels smoothly rotate.

Fabricated robot system is shown in Fig. 2. Moving sequence is programmed in a computer. The computer controls the electromagnetic valves which are 120 m far away through a valve controller, using the air-feeding time schedule program for the actuators. The other ends of air-feeding tubes are attached to the air-compressor and the vacuum pump.

3. Moving principle

The air-feeding time schedule and moving principle of the robot are shown in Fig. 3. The bellows are stretched by positive pressure of 0.1 MPa, and shrunk by negative pressure of -0.08 MPa.

Step 0: All the bellows are in the shrunk condition, because the negative pressure is supplied.

Step 1: The positive pressure is supplied to the bellows 1 and the bellows 1 is stretched. Front end of the bellows 1 moves to the forward direction, because the friction

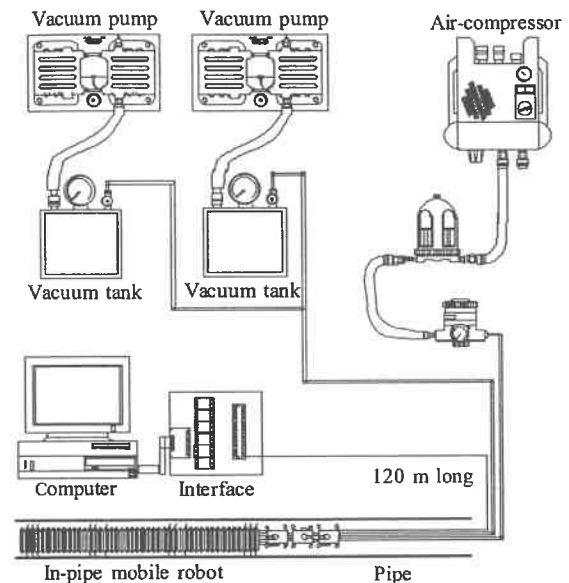


Fig. 2 Experimental apparatus of the mobile robot system

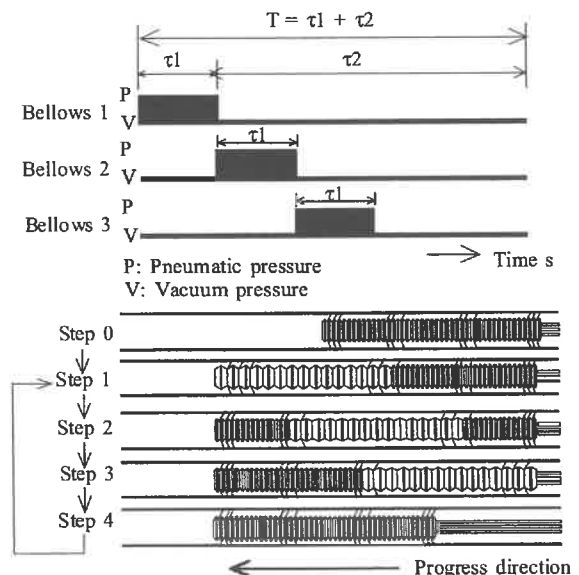


Fig. 3 Principle of the moving of the mobile inspection robot

force of the friction rings at the front end of the bellows 1 is smaller than friction force of the friction rings at the junction between bellows 1 and bellows 2.

Step 2: The negative pressure is supplied to the bellows 1 and the bellows 1

is shrunk. At the same time, the positive pressure is supplied to the bellows 2 and the bellows 2 is stretched. The junction between bellows 1 and bellows 2 moves to the forward direction, because the friction force of the friction rings at the front end of the bellows 1 is larger than friction force of the friction rings at the junction between bellows 1 and bellows 2, beside the friction force of the friction rings at the junction between bellows 1 and bellows 2 is smaller than the friction force of the friction rings at the junction between bellows 2 and bellows 3.

Step 3: The negative pressure is supplied to the bellows 2 and the bellows 2 is shrunk. At the same time, the positive pressure is supplied to the bellows 3 and the bellows 3 is stretched. The junction between bellows 2 and bellows 3 moves to the forward direction, because the friction force of the friction rings at the junction between bellows 1 and bellows 2 is larger than friction force of the friction rings at the junction between bellows 2 and bellows 3, beside the friction force of the friction rings at the junction between bellows 2 and bellows 3 is smaller than the friction force of the friction rings at the end of bellows 3.

Step 4: The negative pressure is supplied to the bellows 3 and the bellows 3 is shrunk. The end of bellows 3 moves to the forward direction, because the friction force of the friction rings at the junction between bellows 2 and bellows 3 is larger than the friction force of the friction rings at the end of bellows 3. One cycle of stretching and shrinking motion of the robot becomes end. The robot can move a stretching distance in the cycle time.

4. Moving experiment

Traction force generated at the friction ring is measured and shown in Fig. 4. The traction force of 52 N is generated at the number of 20 friction rings. Friction force of the air-feeding tubes conveying mechanism in the pipe is measured. Space between each

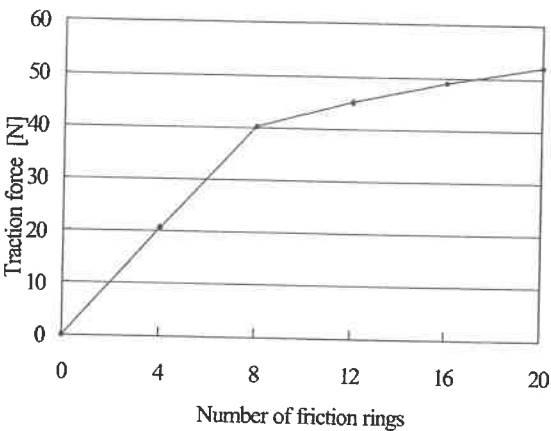


Fig. 4 Relationship between number of friction rings and generated traction force

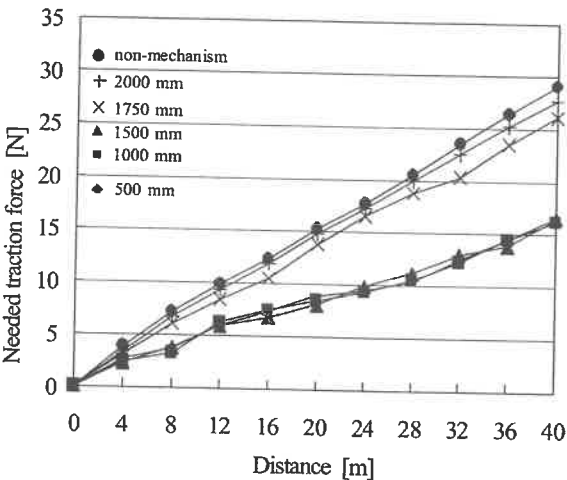


Fig. 5 Relationship between moving distance and needed traction force

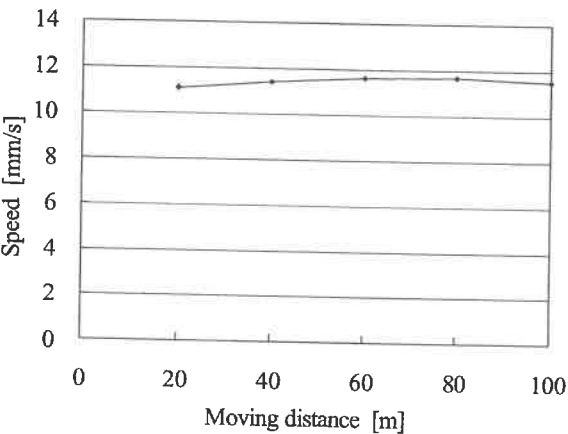


Fig. 6 Relationship between moving distance and speed

mechanism is selected as 500 mm, 1000 mm, 1500 mm, 1750 mm and 2000 mm. Friction force is shown in Fig. 5.

The friction force of the group in 1750 mm, 2000 mm and infinitive (non-mechanism) is about 0.7 N/m, because the air-feeding tubes touch the pipe. The other hands, the friction force of the group in 500 mm, 1000 mm and 1500 mm is about 0.4 N/m, because the air-feeding tubes do not touch the pipe.

So, the space between each mechanism is selected as 500 mm. Then, the friction force of 40 N may be occurred for the moving distance of 100 m. However, the robot equipped 20 friction rings can generate 52 N. Moving speed and moving distance of the robot is shown in Fig. 6. The moving speed increases according to the moving distance. Finally, the moving of 100 m distance was confirmed.

5. Conclusions

(1) We fabricated an artificial earthworm type mobile inspection robot which is constructed by three bellows, 20 friction rings and three electromagnetic valves.

(2) The robot was confirmed to move in the pipe which has 79 mm in inner diameter

and 100 m long.

(3) The traction force depends on the number of friction rings. The inspection robot equipped with 20 friction rings may be able to move more than 130 m. We obtained feasibility on an in-pipe inspection robot of the first cooling-water pipe of atomic power plant.

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Nonlinear Friction Behavior of Discontinuity at Stroke End in a Ball Guide Way

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Abstract: In this paper, the non-linear behavior of friction is modeled with a dynamic system. The non-linear behavior model being proposed here can be used for modeling Coulomb friction, viscous friction, Stribeck effect and can also be used to study the non-linear behavior of elastic deformation. The frictional force does not cause disturbance in the accelerated motion, instead, it lends a support. The proposed model is not symmetrical about the zero-speed of motion. The simulation result demonstrates that the proposed model is more accurate than conventional ones. In order to verify the validity of the proposed model, experimental results are provided. A linear slider is used to perform the friction force compensation experiment to compare the proposed model with a conventional model. The tracking error is seen to have reduced by more than 50% compared to the case when friction compensation is not used. Further, the value of the positive error was sufficiently suppressed compared with conventional models that use compensation.

Keywords: ball guide way, friction compensation, non-linear behavior

1. Introduction

The need for high precision and fast response in NC machines for use in the machine tool industry, especially in the manufacturing of semiconductors, is rapidly growing. In order to achieve high levels of performance, frictional effects have to be addressed. However, only a few methods that address frictional effects have been reported thus far. These methods have failed to consider the dynamic behavior of friction. The non-linear nature of friction gives rise to tracking errors and causes stick-slip motion at low speeds. Friction is a natural phenomenon that is quite hard to model, and it is not yet completely understood. Typical examples of friction are different combinations of Coulomb friction, viscous friction, and Stribeck effect [1,2]. In this paper, the non-linear behavior of friction is modeled with a dynamic system. The non-linear behavior is studied using a one-axis stage that is driven by an AC linear motor and guided by a rolling ball guide [3]. It is shown that simulation results based on the proposed model agree well with experimental results.

2. Nonlinear Frictional Behavior

2.1 Experimental System

The experimental system shown in Fig.1 consists of the following: (i) a one-axis stage mechanism consisting of an AC linear coreless motor which has no cogging and no attraction magnet force, (ii) a rolling guide mechanism (iii) a position-sensor (1 pulse=10nm), (iv) two current amplifiers, and (v) a personal computer with the controller, a D/A board and a counter board. Figure 2 shows a block diagram of the control system, which consists of a P controller (K_p), a PI controller (K_i, T_i), a force filter (T_f), a constant gain (K_f) and a mass (M). FF is Velocity feed-forward gain, CF_r is friction compensation force and F_d is the actual frictional force. The controller is tuned using the normal procedure and the force is calculated by eqn.(1), where V_r is velocity reference, and V_{fb} is velocity response. FF gain is set to zero.

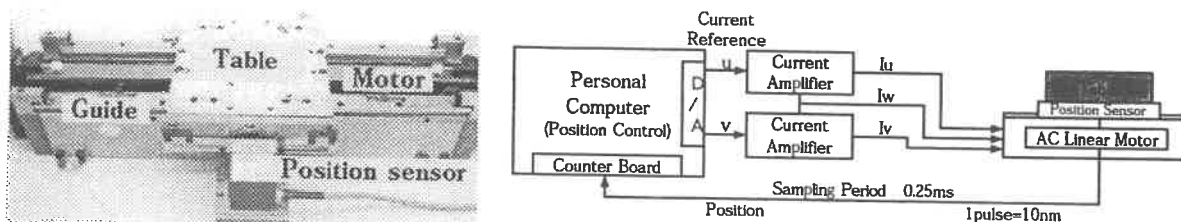


Fig.1: Experimental system.

2.2 Experimental Results and Discussions

In applications requiring high precision positioning and low velocity tracking, the conventional control methods are not always satisfactory. In particular, the tracking error is large at the end of a stroke. The major disturbance source is friction. It is common to regard the friction force as a static function of velocity although more complex friction phenomena using complex models have been proposed and reported. Experiments have shown that there is a deflection or relative movement in the pre-sliding region, indicating that the relationship between the deflection and the applied force resembles a non-linear spring with a hysteretic behavior [4,5].

Thus, the present study focuses on the non-linear behavior at the stroke end during changes in velocity as shown in Fig.3. In Figs 3 and 4, the signals of V_{r1} , V_{r2} , and V_{r3} are velocity references with constant acceleration-deceleration profiles of 2.5mm/s, 5.0mm/s, and 10mm/s, respectively. V_1 , V_2 , and V_3 are velocity responses and F_1 , F_2 , and F_3 are output forces. The forces in the actual experiment are calculated values and not the values mechanically measured. From Fig. 3, it is seen that the output forces are different at zero velocity for different deceleration profiles. When the velocities are decreasing, output forces have not decreased and when the velocities are increasing, output forces have not increased. Further, it was found that the direction of forces changed. Next, the region of decreasing velocities and zero velocities are studied as shown in Fig. 4. In Fig. 4, the output forces decrease when decelerating, and the output forces to keep zero velocities are about 5.3N.

Next, the output forces are set to zero. This causes spring-like behavior in the motion, as shown in Fig. 5. Fig. 5 shows position responses when the output forces are set to zero and the command velocities are 2.5mm/s, 5.0mm/s, and 10mm/s, respectively. At values of low command velocities, the spring-like behaviors produce large displacement (2000pulse=20μm). The displacement, which exceeds 15μm can negatively influence precision point to point control. The frequency of vibration was observed to be 40Hz. The natural frequency does not depend on the magnitude of the frictional forces at zero velocity. The spring-like characteristic behavior is thought to be due to the elastic deformation between balls and rails in the ball guide-way [6].

3. Modeling of Nonlinear Frictional Behavior

3.1 Proposed Friction Model

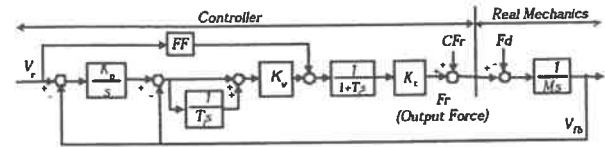


Fig.2: Block diagram of control system.

$$F_r = \frac{K_i}{1 + T_f s} \left\{ K_v \left(1 + \frac{1}{T_i s} \right) \left(\frac{K_p}{s} (V_r - V_b) - V_b \right) + FF V_r \right\} \quad (1)$$

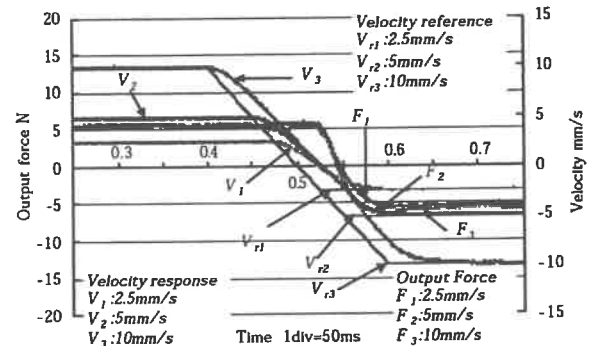


Fig.3: Experimental results at stroke end.

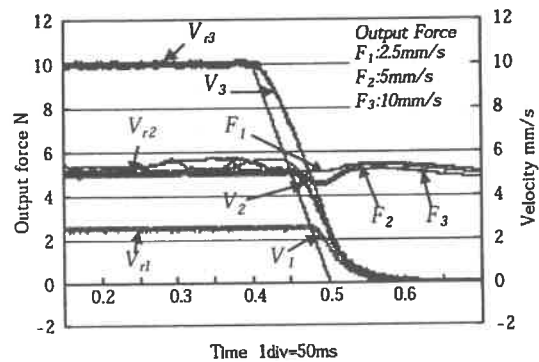


Fig.4: Experimental results at zero velocity.

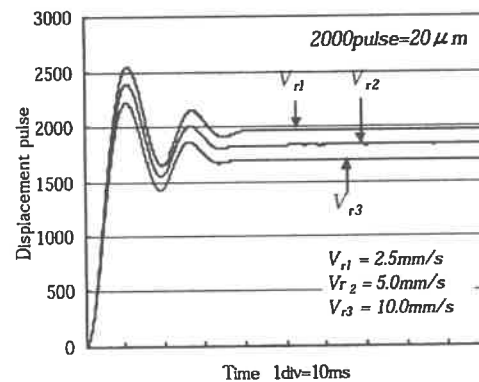


Fig.5: Springlike behavior when output forces are set to zero.

Based on the results of these experiments, a new friction model is developed. The model includes a linear term (viscous friction), a non-linear term (Coulomb friction), which is nearly constant but whose sign depends on the direction of velocity, and an extra component at low velocities (Stribeck effect). During deceleration, the magnitude of the frictional force at zero velocity increases due to Stribeck effect which means the influence of friction increase by direct contact of balls and rails in the mixed lubrication mode at low velocity. When the direction changes and the mass accelerates, the frictional force exhibits spring-like characteristics. The frictional force does not hinder the accelerated motion and in fact lends a support. In this operating region, the present model differs from conventional models, as described below. The curve of friction in the deceleration area and in the acceleration area is not symmetrical about the zero velocity point. It is continuous and is an exponential function, as shown in Fig. 6. Equations are as follows:

$$C_s(t) = -C_1 - C_e \exp((t - t_z)/t_1) - D_s V_{fb} \quad t \leq t_z \quad (2)$$

$$C_s(t) = C_2 - (C_e + C_1 + C_2) \exp((t_z - t)/t_2) + D_s V_{fb} \quad t \geq t_z \quad (3)$$

C_1 and C_2 are Coulomb friction, $C_s(t_z)$ is state force at zero velocity. t_z is the time of zero velocity. The property of the friction model is determined when three parameters, t_1 , t_2 and $C_s(t_z)$ are given.

3.2 Conventional Friction Model

The conventional description of friction is a static relation between velocity and friction force. Tustin's model consists of Coulomb and viscous friction. The inclusion of the Stribeck effect with one or more break points gives a better approximation at low velocities, as shown Fig. 7. Equations for the conventional model areas follows:

$$C_s(t) = -C_1 - C_e \exp((t - t_z)/t_1) - D_s V_{fb} \quad t \leq t_z \quad (4)$$

$$C_s(t) = C_2 + C_e \exp((t_z - t)/t_1) + D_s V_{fb} \quad t \geq t_z \quad (5)$$

The curve has an exponential characteristic and is symmetric about the zero velocity time-axis (t_z). However, it is difficult to define these parameters using linear estimation techniques because of the nonlinear behavior of friction. The three parameters cited above are estimated using GA (Genetic Algorithm) [7]. The proposed friction model's response, the conventional friction model's response and the response obtained experimentally are compared to determine the accuracy of the proposed model, as shown Fig. 8. When the velocity is greater than zero, the estimated force using proposed model is the same as that in conventional model and the real (output) force. When the velocity is changing direction, the force given by conventional model is far from the real force, while the force given by the proposed model is the same as the real force. The proposed model is judged to have good accuracy. Figure 9 shows the estimated frictional force curves as function of velocity. The estimated frictional force curves show that the exponential and continuous curve improves the accuracy of the proposed model. The friction curve during the change in the direction of velocity works as an acceleration force.

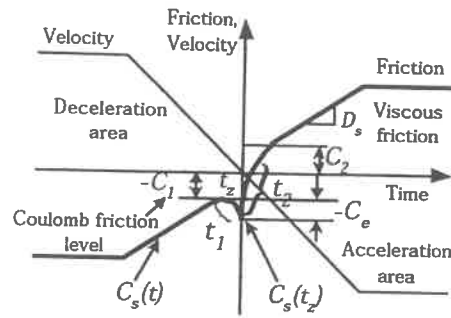


Fig.6: Proposed Friction Model.

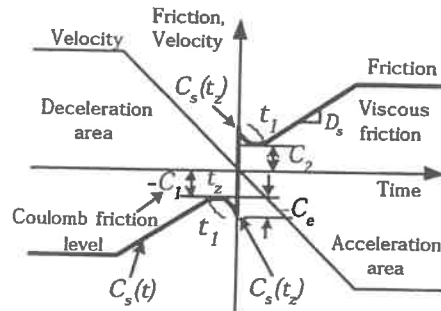


Fig.7: Conventional Friction Model.

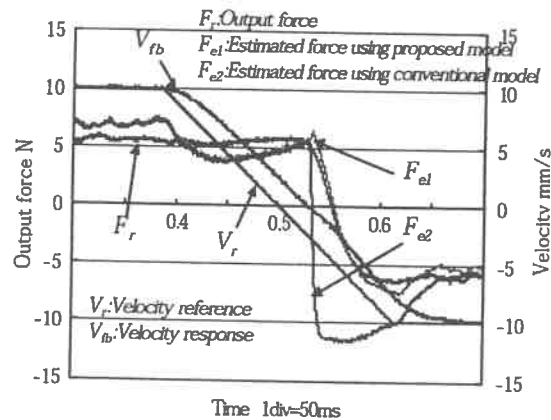


Fig.8: Performance comparison between i. real (output) force, ii. estimated force using proposed model, and iii. estimated force using conventional model at 10mm/s.

4. Compensation of Frictional Behavior

A linear slider is used to perform the friction force compensation experiment using the proposed model and the conventional model. A motion in which the velocity is reversed was used to carry out the experiment. In Fig.10, V_r is velocity reference, while V_1 , V_2 , and V_3 are velocity responses. To reduce the tracking error, FF gain is assumed to be one. P_{e1} is position error without compensation, P_{e2} is position error with compensation using proposed model, and P_{e3} is position error with compensation using conventional model, respectively. The tracking error was reduced to less than a half of that when friction compensation is not used ($-4.7\mu\text{m} \Rightarrow -2.4\mu\text{m}$). When position errors P_{e2} and P_{e3} are compared, positive error occurs in the case of P_{e3} using conventional model ($+1.5\mu\text{m}$). In NC machines, positive error is undesirable, since this would mean that a work piece is excessively cut. Thus, the proposed model is judged to have better performance accuracy.

5. Conclusion

The non-linear behavior of friction is modeled with a dynamic system. Based on simulation and experimental results, the following are the conclusions:

- (1) The non-linear behavior model being proposed here can be used for modeling Coulomb friction, viscous friction, Stribeck effect and can also be used to study the non-linear behavior of elastic deformation. The frictional force does not cause disturbance in the accelerated motion, but lends as a support.
- (2) The simulation demonstrates that the proposed model is much more accurate than conventional ones.
- (3) A linear slider is used to perform the friction force compensation experiment using the proposed method. The tracking error by proposed model was reduced to less than a half of that when friction compensation is not used, and the positive error was sufficiently suppressed compared with conventional model with compensation.

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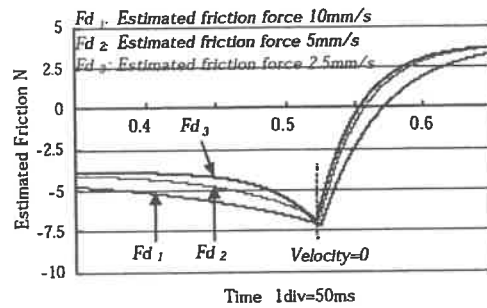


Fig.9: Simulation results of estimated friction force curves using proposed model

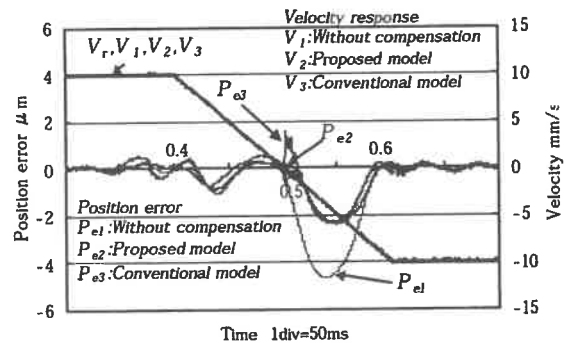


Fig.10: Experimental results obtained for i. Case without friction compensation, ii. Case with compensation using proposed model, and iii. Case with compensation using conventional model at 10mm/s.

GLOBAL TELE-OPERATION FOR MICRO-ASSEMBLY TASKS

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1 INTRODUCTION

New challenges in production technology emerge with the continuing miniaturization of modern products and with the increasing integration of diverse mechatronic components. For prototype assembly and small-lot production, tele-operated micro-assembly systems offer new perspectives concerning ergonomic aspects and process efficiency.

Telepresence and teleaction systems (TPTA) are a promising approach to overcome the scaling barriers between the human operator and the assembly task in microsystem technology (MST). By a TPTA-system an operator can feel physically present in an inaccessible or far-off environment and can carry out manipulations over local or dimensional barriers through the use of suitable tools. In micro production the inaccessibility is mainly caused by the size of the objects to be manipulated and by the required accuracy of the processes [1]. But also clean-room conditions require the local distance of the human operator.

To get a close-to-reality impression of the process conditions, typical TPTA-systems provide haptic input devices, to feel e. g. mounting forces, and user interfaces on the operator side, which visualize the assembly scenario. In this way the operator can control the assembly tools very precisely. Due to the ergonomic working place and the multimodal feedback of relevant process data a tele-operated manual assembly system is more efficient, compared to conventional working environments with microscope and manually triggered tools. Furthermore, new and efficient strategies in global production networks can be realized by using telepresent technologies in conjunction with flexible and modular assembly tools. To investigate these aspects, a tele-operated micro-assembly system has been developed and tested within a transatlantic micro-assembly experiment which is discussed below.

2 SYSTEM OVERVIEW

2.1 Tele-operator end

The tele-operator of the manual micro-assembly system consists of two core components. The first component is an assembly platform, which provides fasteners for part supply equipment, like chip-trays, and for substrates on which the chips are assembled. The platform is attached to a cartesian coordinate axes system with three precision linear tables. With a positioning resolution of $0.1 \mu\text{m}$ and a total workspace of $204 \text{ mm} \times 204 \text{ mm} \times 204 \text{ mm}$ it is suitable for a multitude of assembly processes at a very high accuracy. By a uniaxial force sensor with a measuring range of $\pm 50 \text{ N}$

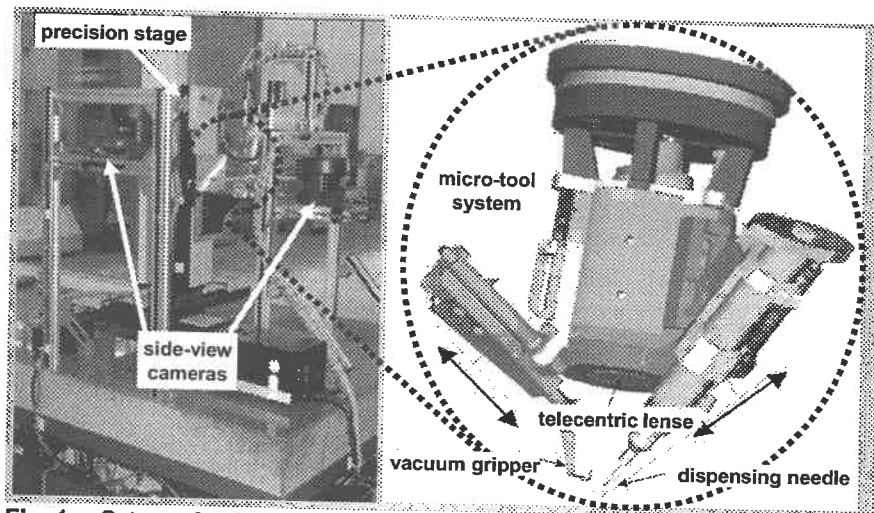


Fig. 1 Setup of tele-operator end: overall view (left) and modular micro-assembly tool head (right)

mounting forces in z-direction can be acquired precisely. In conjunction with a 12 bit data acquisition card, a force resolution of ± 0.024 N can be achieved. This is a suitable resolution for sensitive handling and mounting tasks for microsystems. Several hard- and software safety features are implemented in a multistage safety concept on the tele-operator-end. E. g. to prevent damages to the sensitive force sensor resulting from overloads or collisions, the assembly platform is secured at first by resiliently mounting it to the axis system by using two miniature frictionless spring-loaded tables and electrical limit switches. Also safety algorithms have been implemented into the control application. When exceeding a predefined force limit a further motion of the z-axis is stopped in order to prevent a collision and overload of the sensor. However, retracting motions are still feasible in order to reduce the sensor load and to continue the assembly process. Additionally at predefined safety areas within the working space the maximum speed of the micromanipulator can be reduced in order to minimize the danger of possible collisions.

The second core component, a micro-assembly tool system for tele-operation, is based on the flexible and modular micro-assembly tool head SATURN (sensor-based assembly tool using robot vision) [2, 3], which was developed at the *iwb* for automated micro-assembly tasks (Fig.1, right). An optical system, consisting of a CCD-camera, a highly precise telecentric lens and a coaxial illumination has been integrated in the centre of the tool head. At a working distance of 34.9 mm and a magnification scale of 2:1, the field of view of the mentioned objective amounts to 1.9×2.5 mm and the depth of focus is ± 0.5 mm. For adjusting the optical focus, the whole optical system can be moved within a range of ± 5 mm in the vertical direction. The sensor-based tool head can be rotated by 360° with an angular resolution of 0.005° . Up to four process-specific endeffectors can be flexibly attached onto special interfaces which are arranged equally around the tool head cylinder. The specific tools, e. g. grippers and dispense units, can be extended into the focus plane of the camera by pneumatic cylinders. In summary the tele-operated assembly tool system provides a total number of four degrees of freedom and a total number of four endeffectors, which is sufficient for a multitude of micro-assembly tasks.

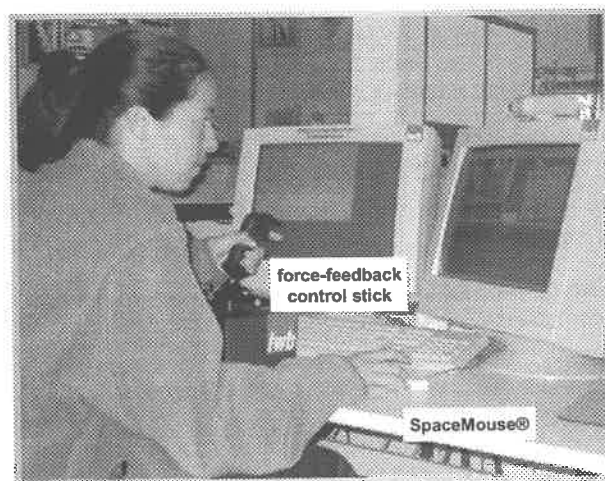
2.2 Operator end

The tele-operated micro-assembly system can be controlled with a haptic input device for micro-assembly operations and a non-haptic input device for auxiliary motions and functions. For achieving a close-to-reality-impression during the manual assembly process, a force-feedback control stick is implemented for controlling the motions of the three precision axes separately. Motions in x and y-direction can be triggered by moving the stick while pushing button 1. The assembly platform can be moved in z-direction by moving the control stick forward and backward while pushing button 1 and 2 together. In conjunction with the mentioned high-resolution force sensor and customized data-processing on the tele-operator side, a close-to-reality impression can be achieved by realistic contact forces.

For auxiliary functions like adapting the view-point and the tool position by rotation, the assembly tool system is driven by a non-haptic input device with six degrees of freedom (SpaceMouse®). Also the adjustment of the focus plane is done by this input device. By pushing its functional keys each endeffector of the assembly tool system can be extended or retracted separately. In addition, specific tool functions, e. g. opening and closing of grippers, can be activated. Visual supervision of the whole micro-assembly platform is achieved by two side-cameras with different viewpoints and zoom functions. Usually the video image of the above mentioned top-view camera is used for the accurate positioning process and the side-view cameras for coarse-positioning.

2.2 Communication platform

Communication between the operator-end and the tele-operator-end is based on an UDP-client-server application in order to be run easily from anywhere by available internet connections. The server



application controls the sensor and kinematic components of the assembly system (tele-operator). All fail-safe features are integrated as closed-loop algorithms into the server program. After processing the sensor-data customized to the operator, the server sends the data like forces, positions etc. to the operator-end by a UDP-socket connection. The control system of the operator-end, e. g. of haptic and non-haptic input devices, serves as the client-application. The achieved sensor-data from the server is directly presented to the user without further processing. The operator-end itself sends the monitored position-data and key-inputs of the input devices to the server-application. At the same time data-streams of the vision-system are sent continuously from the tele-operator-end to the operator-side.

3 TRANSATLANTIC MICRO-ASSEMBLY EXPERIMENT

3.1 Experimental Setup

For evaluation purposes a transatlantic tele-operated micro-assembly experiment was performed (see Fig. 3). The tele-operator-end was located in a clean-room at the research laboratory of the *iwb* in Munich. The tele-operator could be controlled from two different operator stations. The first operator

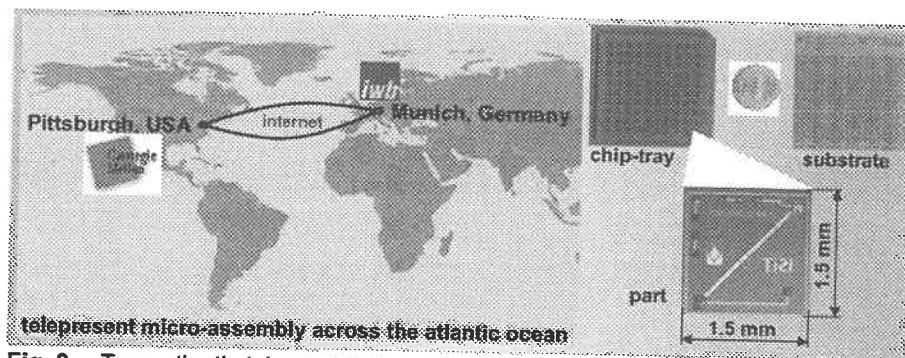


Fig. 3 Transatlantic tele-operated micro-assembly: experimental setup

station was located at the research laboratory of the *iwb* in Munich. The second operator station was located at the Microdynamic Systems Laboratory of the Robotics Institute at the Carnegie Mellon University in Pittsburgh, USA. To communicate verbally and visually, both operators were connected using a video conferencing application.

The task was to pick up an integrated circuit from a chip-tray and place it on a test-circuit board. The dimensions of the integrated circuit were about 1.5 mm × 1.5 mm × 0.5 mm. To show the ability to switch the control between Munich and Pittsburgh during operation, the assembly task was divided in two sub-tasks.

The first sub-task was performed by the operator in Munich. The operator had to pick up the integrated circuit from the chip-tray and to move it into the middle between the chip-tray and the test circuit board. In order to facilitate the navigation, a European 20-Cent coin was placed on the middle between the chip-tray and the test-circuit board.

The second sub-task had to be accomplished by the operator in Pittsburgh. As soon as the integrated circuit was placed above the coin, control was switched to the operator station in Pittsburgh. He then moved the gripper with the integrated circuit above the desired position on the test-circuit board, adjusted the orientation and placed it there. As the top-view video camera provided no information about the distance between the integrated circuit and the substrate, the operator had to rely on haptic information provided by the 1-DOF-force sensor, which was integrated into the teleoperator.

3.2 Results

The experiment has confirmed the capabilities and the applicability of TPTA technologies for manual micro-assembly processes. The average assembly time for performing the above described task was about 3.5 min. Despite the long distance and the resulting time delay between the operator in Pittsburgh and the tele-operator in Munich, no significant detractions of the system performance could be observed. The haptic feedback facilitated sensitive handling of micro-parts very much. E. g., it was possible to stack four integrated circuits exactly upon each other.

For estimating the time delay between the operator-station in Pittsburgh and the tele-operator in Munich the round trip time (RTT) was measured (see Fig. 4). As figure 4 shows, internet data packages are passing through 19 intermediate stations. The measured RTT amounted to about 120 ms. Assuming

In order to estimate the fluctuations of the internet connectivity over a longer time period, another simple experiment was carried out. About 90000 ping-packets

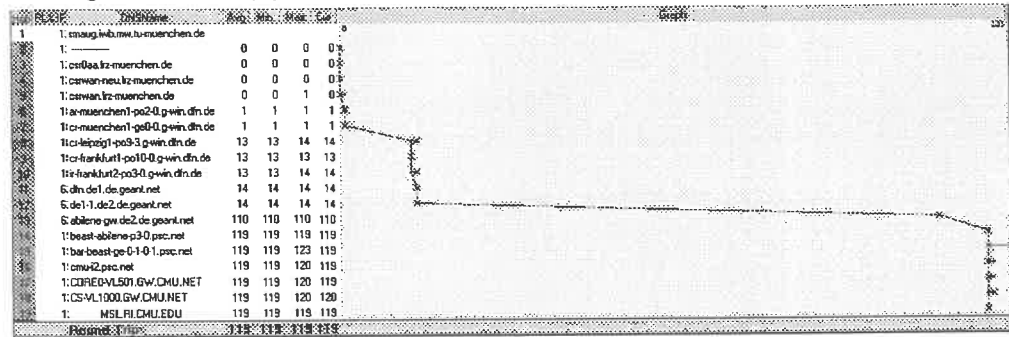


Fig. 4 Traceroute screenshot between Munich and Pittsburgh

4 SUMMARY AND OUTLOOK

Furthermore the experiment shows some interesting options for future industrial applications: By use of TPTA technologies the operator does not have to be physically present in the clean-room, where the micro-assembly system is located. Hence, this enables better working conditions. Furthermore, the workload of tele-operated micro-assembly systems can be improved by shifting to another operator at a different location. For example, short-term lack of staff, e.g. due to illness, holidays, strikes etc., can be easily compensated. Also, cost-effective 24 h production can be achieved by shifting to operators in different time-zones.

5 ACKNOWLEDGEMENTS

6 REFERENCES

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Multi-Axis Nano Positioning Flexure Stage Using Magnetically Preloaded PZT Actuator

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Abstract

This paper presents a novel 3-axis fine positioning stage using magnetically preloaded PZT actuators. All the procedure concerning the design, fabrication, experiment and performance evaluation of the stage are described. The stage considered here is composed of flexure hinges, piezoelectric actuators and their peripherals. A special flexure hinge is adopted to actuate the single stage in three axes at the same time. The developed stage makes use of magnetically preloaded PZT actuators and a ball contact mechanism to avoid the cross-talk among the axes. The structural design for the stage is analyzed and optimized with the help of a finite element computer simulation. Performance evaluation is also made for the PZT actuators as well as the stage positioning accuracy. Experimental results show that the developed stage is accurate enough to be used for nanometer positioning.

1. Introduction

With the increasing demand on more precise and higher speed stages, nano positioning stages have attracted much attention. Recently, several types of PZT driven nano positioning stages have been developed to assess more rapid response and higher precision. In particular, many researchers and industry are concerned with multi-axis nano positioning stages because of their usefulness in semi-conductor manufacturing systems and scanning apparatus [1,2].

This paper introduces a novel multi-axis stage using magnetically preloaded PZT actuators. The developed stage has various advantages in implementation for multi-axis positioning systems: the magnetically preloaded PZT actuators allow longer stroke; the motion crosstalk is minimized by using a ball contact mechanism between the moving part and the fixed part; the proposed stage is realized to be more compact than other stages by realizing 3 axis motion mechanism based upon a single moving stage.

The structural design for the stage is analyzed and optimized with the help of a finite element computer simulation. Experiments are also performed to investigate the performance of the developed, 3-axis nano positioning stage using magnetically preloaded PZT actuators. Throughout the experiments, it is validated that the proposed stage has good performance in accuracy.

2. Design, Modeling and Analysis of Stage

Figures 1 and 2 show the actual stage and the zoomed view of flexure hinge used. Figure 3 is a PZT actuator assembly to drive stage, in which an upper flux plate is installed at the stage moving part and a permanent magnet is employed to provide magnetical preload. The PZT stack, which has the length of 40 mm, is capable of producing maximum stroke of $30\text{ }\mu\text{m}$ with the open-loop resolution of 0.3 nm. The push and pull force capacity are 1000N and 5N, respectively. Since the PZT stack is fragile under tensile loading condition, it is important to preload the stack in order to prevent any damage from dynamic tensile loads. It is recommended [1] that the preloading force be around 20 percent of the maximum force capacity to preserve sufficient stroke, while protecting the stack from damage. For magnetic preloading, permanent magnets made of Nd-Fe-B are adopted. The outer and inner radii of the permanent magnets are 15mm and 6 mm, respectively, and the axial length of the magnets is 25mm. Figure 4 shows a ball($\phi=2\text{mm}$) joint between the plate and PZT actuator. The purpose of the ball joint is to eliminate motion crosstalks. Flexure hinge is adopted to actuate the single stage in three axes at the same time. A numerical analysis based on FEM is made to obtain best parameters for flexure hinge.

3. Fabrication and Preliminary Experiment

The characteristics of the developed stage is summarized in Table 1. The stage is made of aluminum and the magnetical preloading system makes use of a permanent magnet. A preliminary experiment is made to verify the design procedure based on FEM modeling and formula. Figure 5 shows the measured frequency response function, in which the measured and computed natural frequencies are indicated. The figure confirms that the proposed analysis method is accurate enough to be used in the design procedure. On the other hand, the magnetic preload can affect the dynamic characteristics of the stage. Figure 6 shows the change of natural frequencies with the variation of the preload given to each moving axis. It can be concluded that the preload increases the natural frequency. This is believed to be due to the ball contact mechanism, of which stiffness is significantly affected by the preload [3].

4. Performance Evaluation

To evaluate the developed stage, maximum strokes and resolutions about input voltage(0~120V) are measured. An experiment to investigate the decrease of maximum stroke with magnetically preload is also made. The open-loop characteristics of the magnet-preloaded actuator are measured and compared to those of the spring-preloaded actuator. Figure 7 shows the input-displacement curves with the magnetically preload changed. It is clearly shown that the magnetically preload does not significantly affect the stroke. Figure 8 illustrates the step responses for the stage. Figure 9 shows step tests with 10nm resolution for X and Y axis and with $0.012\text{ }\mu\text{rad}$ resolution for Θ_z axis. The results of stroke and resolution of stage verifies the developed stage is useful as a nanometer positioning system.

5. Concluding Remarks.

In this paper, a novel 3-axis fine positioning stage with nanometer resolution is presented. The developed stage features a single module based multi-axis positioning system. The stage employs the magnetical preload. The design procedure is also presented and verified through an experiment. The accuracy of the developed stage is evaluated with step tests. The experimental results show that the presented stage may be useful for precision equipment.

Acknowledgment

This research was financially supported by “The National Program for Development of Advanced Machinery and Parts”, Korea.

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Table 1 Specifications for stage

Category	Item		Data
Stage	Material	Type	Al 7075-T6
		Density	2810 kg/m ³
		Young's modulus	72 GPa
	Mass of moving part	Without accessories	0.796 kg
		With accessories	0.977 kg
PZT actuator	Type		PSt 150/7×7/20
	Max. Stroke		30 μm
	Stiffness		120e6 N/m
	Preload		Permanent magnet

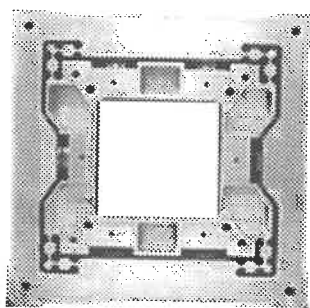


Fig. 1 The developed fine stage

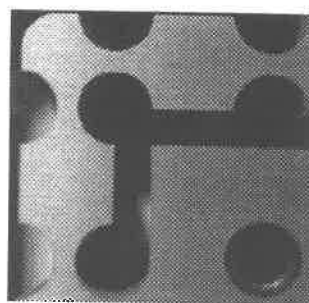


Fig. 2 The flexure hinge

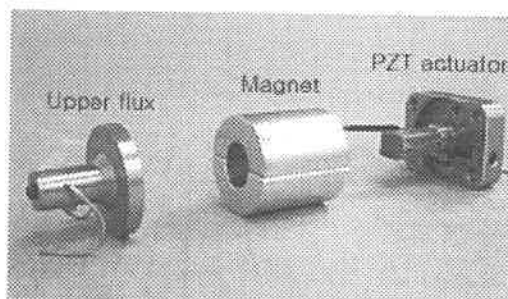


Fig. 3 The PZT actuator assembly



Fig. 4 Ball joint with φ 2 mm ball

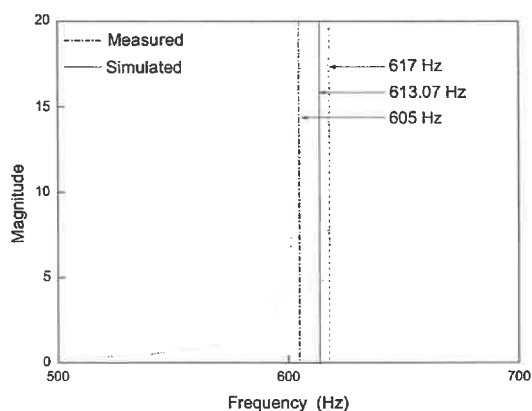


Fig. 5 Frequency response and the natural frequencies of stage moving part without accessories

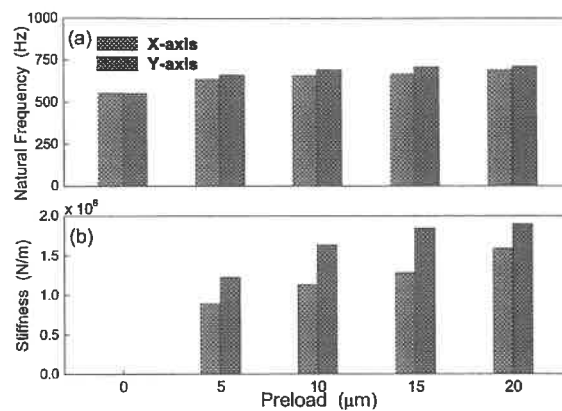


Fig. 6 Changes of the 1st natural frequency and added stiffness with the preload varied

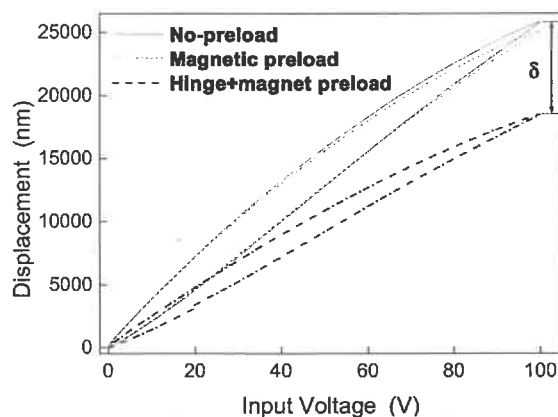


Fig. 7 Comparison of hysteresis loops with and without preloads

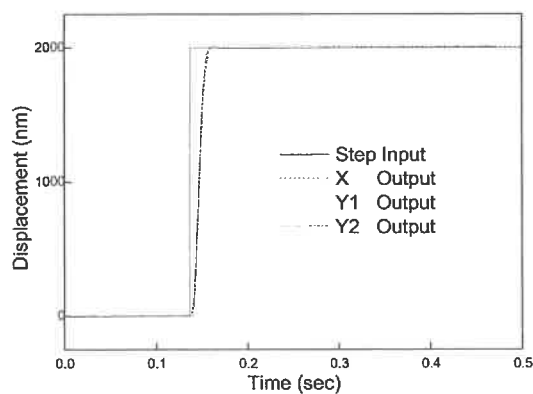


Fig. 8 Step responses

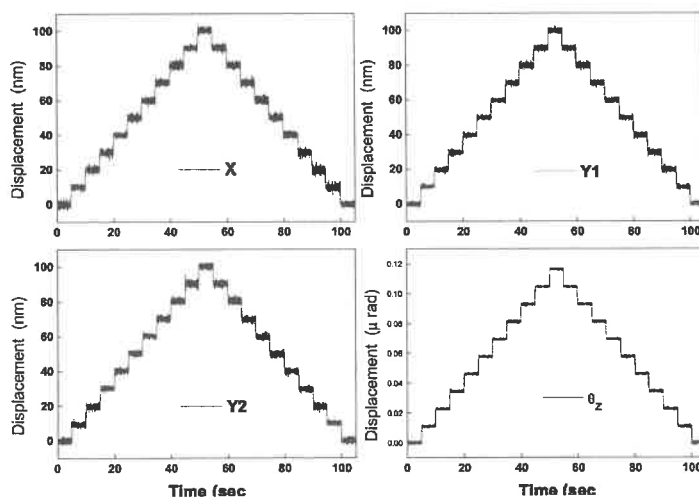


Fig. 9 Step tests with 10 nm and 0.012 μ rad resolutions

DEVELOPMENT OF A MICRO SCREW THREAD AND A MICRO GEAR UTILIZED EXTRA FINE WIRES

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1. Introduction

Micro-mechanical parts for a micro-machine are mostly produced by two methods of a photolithography¹⁾ 2) or an ultra-precise milling³⁾. If we choose the photolithography method to make the micro parts, we can produce a lot of the micro-mechanical parts at a time like IC manufacturing; also, the parts are fabricated into the micro-machines through the processes of multi-layered deposition/electroforming and etching of sacrificial layers. However, the parts are shaped into a planar figurer with constant thickness, and their materials are selected from a few materials example of poly-silicon or tungsten. Moreover, the method requires photolithography equipments for IC manufacturing systems; besides, their equipments must be kept in a clean room. The price of the lithography equipments is over several million dollars and running cost for the clean room runs up to considerable sum. Therefore, the researchers working on the development of the micro-machines which parts are produced by the photolithography method are only a handful of researchers in a few research institutes.

If we choose the ultra-precise milling method to make the micro parts, we can produce only one micro part in one manufacturing cycle; also, the part is not fabricated into the micro-machine. Nevertheless, the micro part is shaped into three dimensional shape by CNC (computerized numerical control) machine tool with CAD/CAM system, and their materials are selected from many materials. However, the method requires a super-precision CNC machining center to enable to control five-axis machining simultaneously; besides, the CNC machining center needs diamond tools which cutting edges are fabricated to special profiles with the accuracy of sub-micrometer. The price of the super-precision CNC machining center is over several million dollars. Therefore, advanced researches for the micro-machine which parts are produced by the ultra-precise milling method are carried on the shoulders of limited researchers.

Many researchers and engineers wait for the third method to developed micro-mechanical parts with simple and inexpensive processes and without special equipments. A novel method to make the micro mechanical parts utilized extra fine wires is thought up in this paper. A micro-screw, a micro-nut, and an external micro gear, an internal micro gear utilized extra fine wires are developed in this paper as prompt results.

2. Micro screw utilized extra fine wires

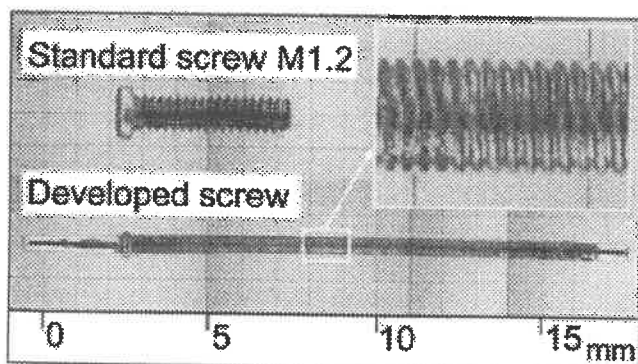


Figure 1. Micro screw utilized extra fine wires and the standard screw of M1.2.

The micro screw utilized extra fine wires and the standard screw of M1.2 are shown in figure 1. The micro screw is made by brazing an extra fine wire which is coiled around a needle pin. The micro screw has following dimensions; its major diameter is 0.6 millimeters, its pitch is 0.1 millimeters (i.e. 100 micrometers), and its length of threaded portion is 14 millimeters. Just for reference, the standard screw of M1.2 has following dimensions; major diameter is 1.2 millimeters, its pitch is 0.25 millimeters (i.e.

250 micrometers), and its length of threaded portion is 6 millimeters. Although the minimum screw thread of S0.3 standardized by ISO/R 1501:1970 has nominal diameter of 0.3 millimeters and pitch of 0.08 millimeters (i.e. 80 micrometers), the author has not been obtained the minimum screw thread of S0.3. The minimum screw thread of S0.3 may not have been manufactured. A pair of a miniature screw of M1.2 and a miniature nut of M1.2 is obtained commercially as a pair of the minimum screw and the minimum nut. ISO/ R 1501:1970 is same as JIS B 0201:1973. An enlarged photograph of the micro-screw utilized extra fine wires is shown in figure 2.

3. Making processes for the micro screw utilized extra fine wires

The micro screw utilized extra fine wires is made through following processes as shown in figure 4. Firstly, a needle pin to be firmly soldered is prepared. We used a terminal pin of a hybrid IC as the needle pin. The pin is undercoated with nickel plating and is plated with gold. Its diameter is 0.5 millimeters and its length is 15 millimeters. Secondly, a pair of one extra fine wire to be firmly soldered on the needle pin and another extra fine wire to be hardly soldered on the needle pin is closely coiled each other around the needle pin like a double-start thread. We used a tinning copper wire as the extra fine wire to be firmly soldered and used a tungsten wire as the extra fine wire to be hardly soldered. Both diameters of the tinning copper wire and the tungsten wire are same size of 50 micrometers. The pair of the wires is coiled by hand. Thirdly, solder is melted upon the double-coiled part mentioned above, and the melted solder flows into gaps between the needle pin and the coiled wires. The extra fine wire to be firmly soldered is brazed around the terminal pin, yet the extra fine wire to be hardly soldered is not brazed around the needle pin. We used the solder which consists of 50 percent tin (Sn) and 50 percent lead (Pb). Finally, the wire to be hardly soldered is removed, and the extra fine wire to be firmly soldered is remained as a ridge of the external thread. Axial cross section of the micro screw shapes a profile to connect semicircle concave ditches and semicircle convex ridges alternatively as shown figure 4. We removed the tungsten wire by hand.

4. The micro nut utilized extra fine wires

The micro nut utilized extra fine wires and the standard nut of M1.2 are shown in figure 5. The nut utilized extra fine wires is made by brazing an extra fine wire which is coiled on the inside of a small tube. The micro nut has following dimensions; its major diameter is 0.65 millimeters, its pitch is 0.1 millimeters (i.e. 100 micrometers), and its thickness is 4

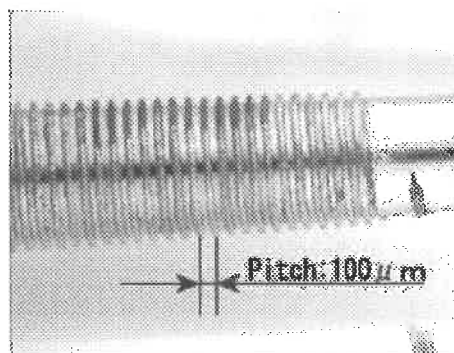


Figure 2. Enlarged photograph of the micro screw.

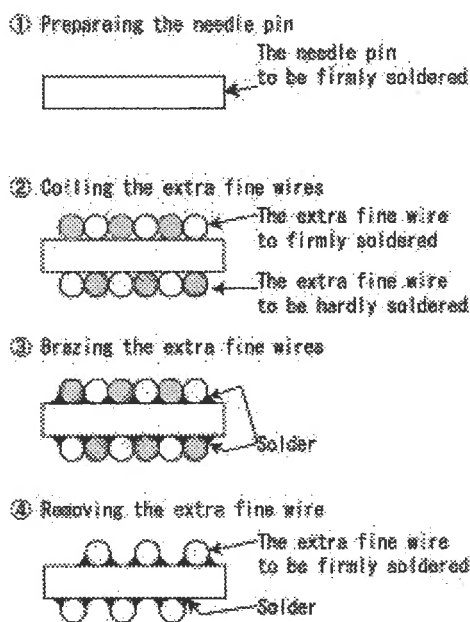


Figure 3. Making processes for the micro screw.

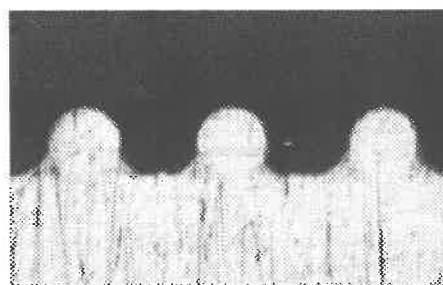


Figure 4. Axial cross section of the micro screw.

millimeters. Just for reference, the standard nut of M1.2 has following dimensions; its major diameter is 1.2 millimeters, its pitch is 0.25 millimeters (i.e. 250 micrometers), and its thickness is 1.2 millimeters. An enlarged

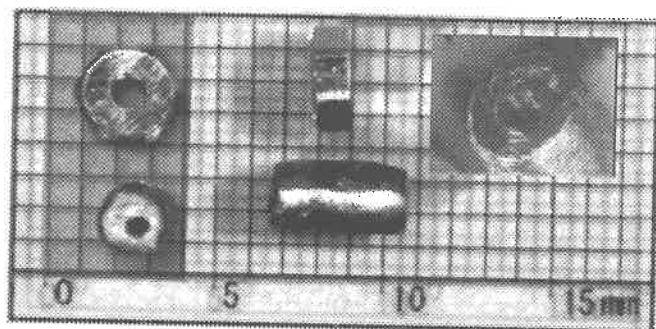


Figure 5. The micro nut utilized extra fine wires and the standard nut of M1.2.

photograph of the micro nut is shown in figure 6.

5. Making processes for the micro nut utilized extra fine wires

The micro nut utilized extra fine wires is made through following processes as shown in figure 7. Firstly, a needle pin to be hardly soldered is prepared. We used a hypodermic needle as the needle pin. The pin is made of stainless steel and is hardly soldered. Its diameter is 0.55 millimeters and its length is 15 millimeters. Secondly, a pair of one extra fine wire to be firmly soldered and another extra fine wire to be hardly soldered is closely coiled each other around the needle pin like a double-start thread. We used a tinning copper wire as the extra fine wire to be firmly soldered and used a tungsten wire as the extra fine wire to be hardly soldered. Both diameters of the tinning copper wire and the tungsten wire are same size of 50 micrometers. The pair of the wires is coiled by hand. Thirdly, the double-coiled part mentioned above is inserted in the small tube, and the melted solder is poured into gap between the double-coiled part and the small tube. The extra fine wire to be firmly soldered is brazed on the inside of the small tube, yet the extra fine wire to be hardly soldered and the needle pin are not brazed. We used an electric sleeve as the small tube. The sleeve is made of copper and is plated with tin inside and outside. Its external diameter is 2.1 millimeters, and its internal diameter is 1.1 millimeters. Its length is 4 millimeters. The solder consists of 50 percent tin (Sn) and 50 percent lead (Pb). Fourthly, the needle pin is drawn out from the brazed part mentioned above. Finally, the wire to be hardly soldered is removed through the center hole which is made by removing the needle pin. The extra fine wire to be firmly soldered is remained as a ridge of the internal thread. Axial cross section of the micro nut shapes the profile to connect semicircle concave ditches and semicircle convex ridges alternatively as shown in figure 8. We removed the hypodermic needle and the tungsten wire by hand.

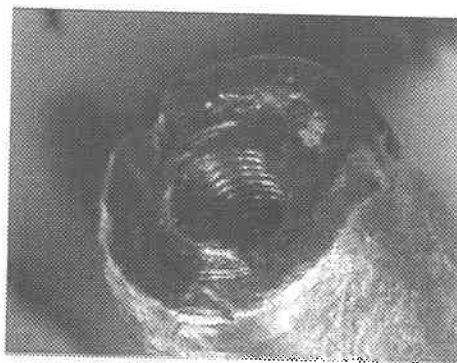


Figure 6. Enlarged photograph of the micro nut.

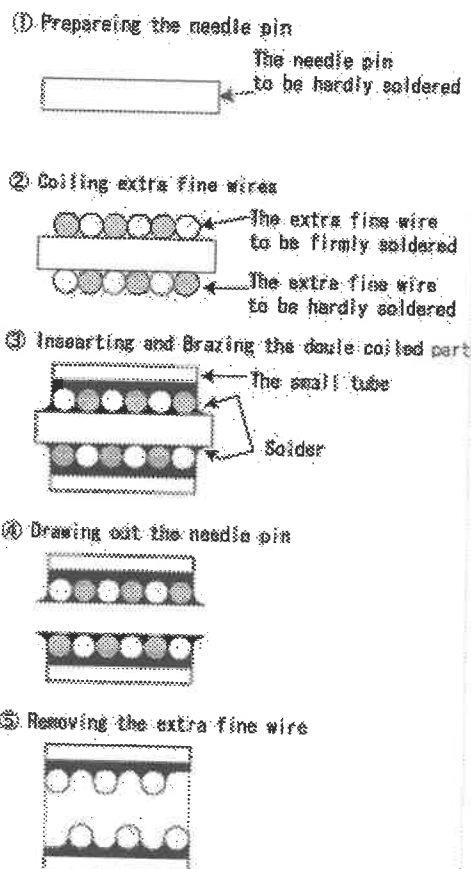


Figure 7. Making processes for the micro nut

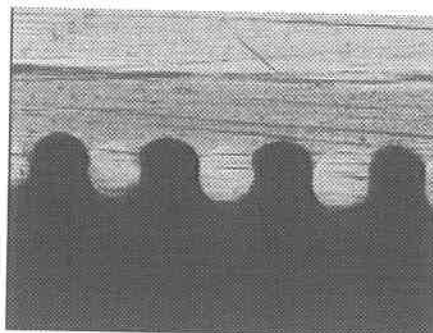


Figure 8. Axial cross section of the micro screw

6. Pitch error of the micro screw utilized extra fine wires

Pitch of the micro screw utilized extra fine wires is measured by a surface roughness measuring instrument with diamond stylus. Axial profile of the micro screw is measured as shown figure 9 because the bottom points in the figure are recorded when both slopes of stylus tip are in contact with surfaces of adjoining the ridges of the micro screw. Distance between adjoining bottom points indicates the pitch of the micro screw. From the result of the measurements, the maximum pitch was 103 micrometers, the minimum pitch was 100 micrometers, and the average of pitch was 100.4 micrometers.

7. The external micro gear and the internal micro gear utilized extra fine wires

The external micro gear and the internal micro gear utilized extra fine wires are developed as shown in figure 10 as prompt results. We could rotate the micro gears engaged each other with angular velocity of 1000 rpm. The external micro gear and the internal gear have a same module of 0.127 millimeters (i.e. their pitches are same length of 400 micrometers), and the number of teeth of the external gear and the internal gear are 9 teeth and 11 teeth respectively.

8. The screw drive mechanism by using the micro screw and the micro nut

The screw drive mechanism in which the micro nut travels 60 micrometers by one turn of the micro screw is developed as shown in figure 11. An angle plate is developed as shown in figure 11. An angle plate is connected to the micro nut for restricting rotation. The micro screw and the micro nut are manufactured by using the extra fine wires of 30 micrometers in diameter, and both pitches are same length of 60 micrometers.

9. Conclusions

The micro screw and the micro nut with pitch of 100 micrometers are developed by utilizing extra fine wires of 50 micrometers in diameter. The external micro gear, the internal micro gear, and the screw drive mechanism are manufactured as prompt results.

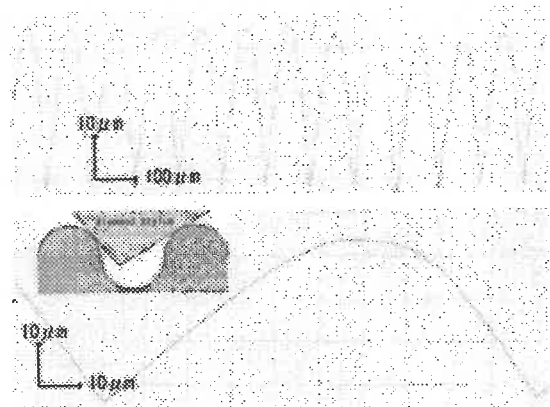


Figure 9. Axial profile of the micro screw measured by the surface roughness measuring instrument.

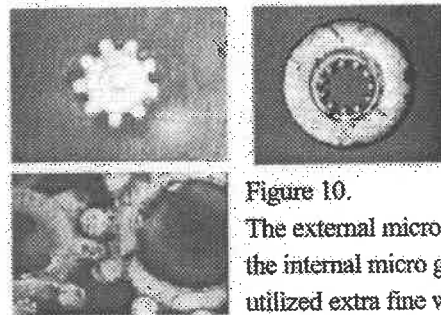


Figure 10. The external micro gear and the internal micro gear utilized extra fine wires.

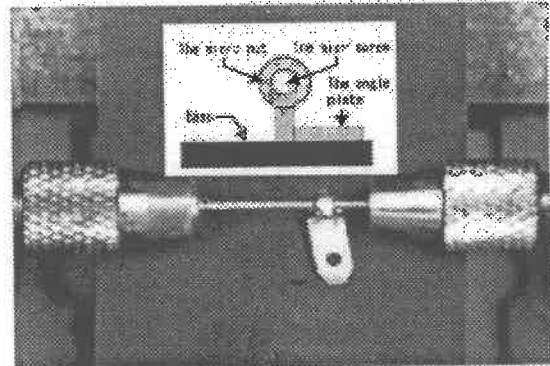


Figure 11. The screw drive mechanism

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A Dual Stage Compliant Positioning System with Integrated Planar Optical Grating for Position Feedback

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Abstract

Assembly, handling and dispensing systems for small-scale parts play an essential part in reducing the cost of these products. These operations require both long coarse positioning for transfer between operations as well as dispensing as well as fine short range positioning on the submicron level for feature assembly. Typical prismatic rolling guide-way based positioning stages with ballscrew drives are to inaccurate for positioning in the submicron level which led to the development of dual stage positioning systems [1]. One stage performs the coarse long-range motions whereas another stage provides small precise motions. One disadvantage of such systems is that each actuator requires individual motion feedback, which leads to higher system cost. Focus of this paper is the design of a positioning stage that uses a dual stage approach for coarse and fine positioning, but utilizes a single two-dimensional optical grating interferometer as position feedback for the resultant motion feedback. This results in significant reduction in error sources and measurement uncertainties due to misalignment and measurement system errors.

Keywords: Micro assembly; Optical position feedback; Friction drive

Introduction

Positioning stages with nanometer accuracy can be designed using compliant mechanisms and piezo actuators, however their travel range is very limited. Therefore if the travel range goes beyond several millimeters a dual approach is pursued. However this approach is much more costly since it requires additional sensors and actuators. The use of individual displacement sensors for each stage leads to an increased error budget. In addition a single reference coordinate frame is not available which decreases the repeatability of the system [2, 3]. This paper introduces a combination of a friction drive for the long range travel, and a conventional flexure stage for the perpendicular travel. Unique to this approach is that a single two degree of freedom optical encoder with a grating scale is used. The longer-range positioning stage with a motion range of 200mm is a conventional linear guide way system. It has an optical grating mounted in the center neutral plane of the stage (Figure 1). This prevents the influence of bending forces on the optical grating. The optical grating has an optical resolution of 4 nm by means of interpolation of the 4 micrometer grating pitch. The optical grating is rigidly attached along the guide-way to provide a reference coordinate system.

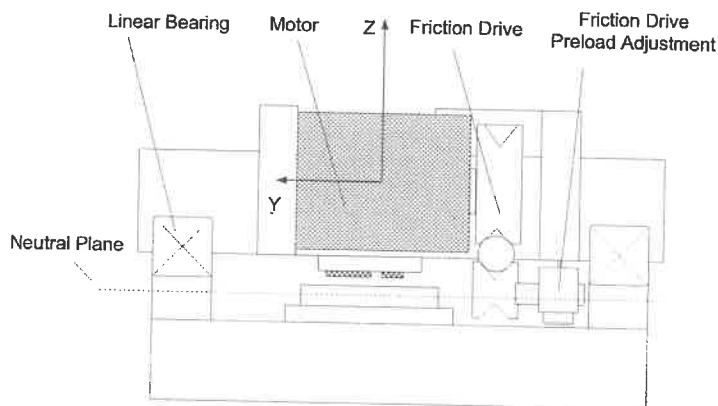


Figure 1: Schematic of the friction drive.

The drive system consists of a vee roller friction drive [4] that allows for very fine positioning. The friction drive has a driven stage that includes the motor and a stationary drive rod for traction. This configuration has the advantage of smaller footprint and the possibility of multiple drives on one stage for parts transfer. A disadvantage is the increased inertia of the moving system, which has an influence on the acceleration and deceleration curve of the drive. However once the system is tuned the system performs robust since the payload is typically small compared to the small scale part weight.

To increase the accuracy and repeatability of the system the drive system is decoupled via a flexure, and directions other than the driven direction are weak (Figure 2).

The second stage for small precise motions of up to 1mm in y direction consists of a flexure system for slip stick free motion. This off the shelf flexure stage has a two direction sensing optical encoder attached to the moving platform that measures the relative position of the flexure stage versus the optical grating along the guide way in x and y direction.

Friction Drive Design

The vee roller friction drive is sized along the following requirements:

- Maximum displacement precision dL for the encoder resolution dE and the wheel Radius R
- Limiting acceleration a based on mass m , the friction μ , and the preload P
- The desired and achievable rectilinear velocity V
- The tangential stiffness of the drive F_t based on the mindlin formula
- Maximum allowable contact stress σ dependent on the contact area A

$$dL = 2 \cdot dE \cdot \pi \cdot R \quad (1)$$

$$a = \frac{F \cdot \mu}{m} \quad (2)$$

$$V = R \cdot \omega \quad (3)$$

$$\sigma = \frac{F}{\pi \cdot A} \quad (4)$$

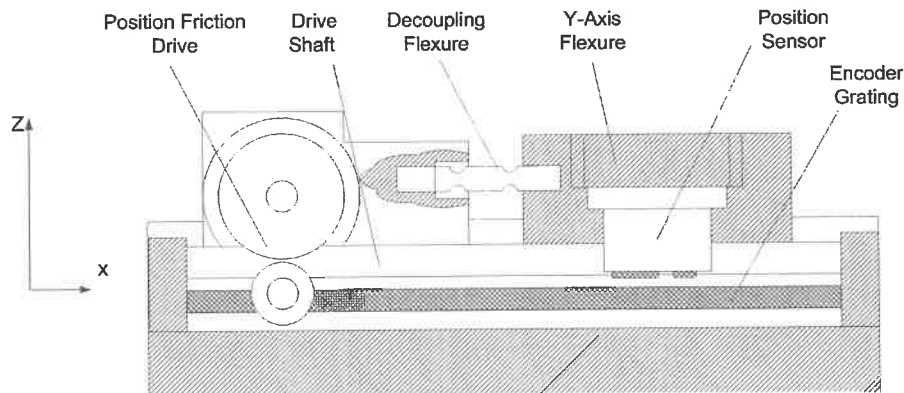


Figure 2: Schematic of the friction drive and the positioning unit.

Connecting Flexure

The connecting flexure is designed to provide a bending stiffness that is lower than the bearing stiffness of the driven slide.

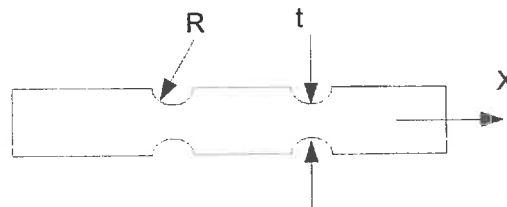


Figure 3: Parameters of the compliant link.

The compliant link stiffness can be calculated for axial, torsion and bending stiffness:

$$\text{Axial Stiffness: } k_a = \frac{Et^{3/2}}{20R^{1/2}} \quad (5)$$

$$\text{Torsion Stiffness: } \phi = \int_0^L \frac{Tdx}{GI_{px}} \quad (6)$$

$$\text{Bending Stiffness: } k_b = \frac{Et^{7/2}}{20R^{3/2}} \quad (7)$$

The parameter G is the shear modulus, T is the applied moment and I_{px} the polar moment of inertia that varies over the length L

Position Feedback and Error Compensation

The Grating of the planar encoder with a length of up to 200mm and a width of 20mm is glued on the stainless steel stage surface. Since the stage is assembled and operated in a controlled environment the thermal behavior of the is stable. The heat generated by the motor is decoupled via the compliant link. The optical read head is mounted to the flexure stage. The errors due to mounting of the grating and of the optical grating can be greatly reduced if a laser interferometer is used. The straightness and translation errors of the stage are measured and compensated via a position dependant polynomial scheme for off axis errors and a lookup table for on-axis errors that resides in the control system [4]. Grating encoders as well as laser interferometers use two sinusoidal waveforms as out put of their photo detectors as representation of position and direction. This signal is commonly 1 Volt Peak to Peak. The following errors can occur in this representation:

- A DC offset in either sinusoidal waveform DC
- A phase offset Ph
- A deviation in the amplitude between both signals.
- A shape error of the signal
- An error in the actual grating pitch

Typically the shape of the sinussoidal signals is represented in a lissajou figure in which a dc offset will result in a center offset of the figure in either x or y direction and a phase error will result in an inclined circle. If the amplitude relationship of both signals is off an "egg" shape results. Heydemann et al presented an software compensation scheme, which has been implemented in commercial systems. But these calculations lower the sampling time of the data acquisition. Therefore in this system the electronics is tuned to provide a high resolution. Table one shows allowable interpolation error values for the grating with 4micrometer grating pitch as target. Please note that these errors are not accumulative.

Error Source	Error in Degree	Error in %	Interpolation Error [μm]
Amplitude mismatch		2	0.006
Phase error	0.5		0.003
Shape error		0.1	0.004
DC offset	0.5		0.003
Resulting error			0.010

Table 1 shows the adjustment specification for a interpolation error of less than 10nm.

Figure 4 shows the output signal of the encoder, and its resulting lissajou figure. The values are based on the error values in column 2 and 3 of table 1, and the resulting interpolation error in micrometer.

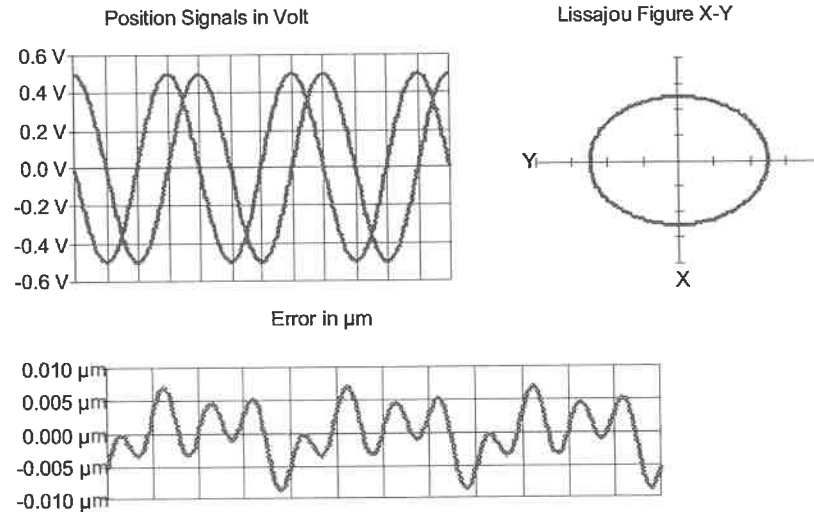


Figure 4: The lissajou graph and the interpolation error caused by signal errors of table 1.

Control Concept

The control uses conventional PID servo control for both axis, however an x-axis lookup table for enhanced accuracy is added as well as the polynomial compensation as function of the traveled length x is added. In order to avoid slip the acceleration curve is adjusted and the torque command is limited below the slip torque.

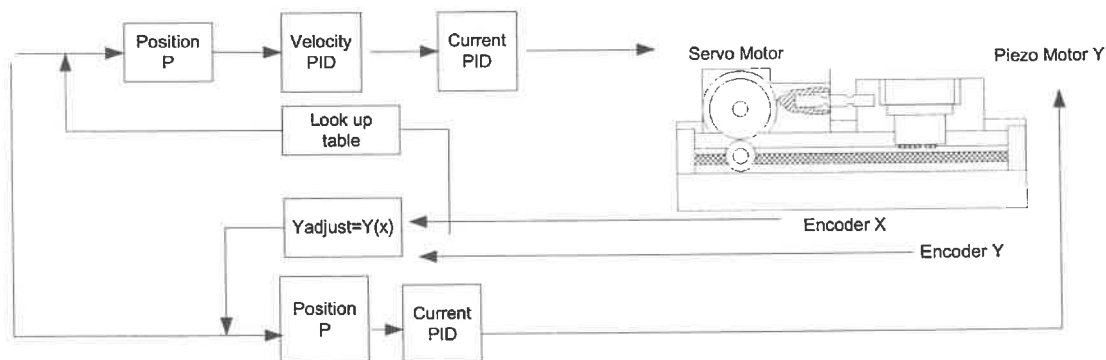


Figure 5: The control schematic for the stage

Conclusion

The design approach for a dual positioning friction drive stage was shown and the associated feedback system error compensation was introduced. A control method for reliable high precision positioning was also presented. This design approach will be verified in a future laboratory study on the finalized Stage.

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The Application of Surface Mount Technology to Multi-Scale Process Intensification

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ABSTRACT

Microtechnology-based Energy and Chemical Systems (MECS) are multi-scale microsystems, which rely on embedded nano and micro-scale features to process bulk amounts of fluids in applications ranging from man-portable heat pumps, distributed fuel reforming and in-situ waste remediation. Microlamination is a well-known process architecture for fabricating the massively parallel, high aspect ratio microchannel arrays needed within these devices. While the promise of MECS technology is exciting, the dissemination of MECS technology is greatly dependent upon economical fabrication. This paper explores the application of surface mount technology (SMT) from the electronics industry to the microlamination of microchannel arrays. An economic analysis shows the impact that SMT could have on the cost of MECS technology. Experimental results show that the fabrication of high-aspect-ratio microchannels is indeed feasible.

Keywords: microchannel arrays, high aspect ratio, surface mount technology, soldering, process intensification

INTRODUCTION

Microtechnology-based Energy and Chemical Systems (MECS) are multi-scale fluidic devices, which rely on embedded micro-scale features to process bulk amounts of fluids in applications ranging from man-portable heat pumps, distributed fuel reforming and in-situ waste remediation. Many MECS devices are fabricated using a process architecture known as microlamination. Microlamination involves three steps: (1) lamina patterning, (2) laminae registration and (3) laminae bonding.¹ Lamina patterning involves surface machining, or through cutting, thin sheets of material, or laminae, with the appropriate patterns necessary to produce the overall desired structure. The laminae are shims or films of a base material (metal, polymer, etc.) having desirable mechanical, thermal and chemical properties important to the functioning of the final device. Eventually the laminae are registered relative to each other and bonded together in a stack.

Many bonding techniques have been used in the past for producing MECS devices. Electron beam welding was used by Beir² to join the laminae of a microchannel heat exchanger. This is a very expensive and time-consuming process. Diffusion bonding (DB) has been widely used in microlamination.^{3,4} This method is also time consuming and requires high temperature and pressure. Such extreme conditions can produce unwanted warpage and residual stress in materials leading to geometric variations and misalignment between laminae. Other techniques used for bonding have included diffusion soldering and brazing.⁴ All of these methods require extreme operating parameters, vacuum conditions and/or long cycle times. Under these bonding regimes, it has been found that the bonding process not only controls the amount of shape variation within microchannels but it also dominates the cost in microlamination.⁵

A key impediment to the proliferation of MECS applications will be its economical production. It is expected that with the diversity of MECS products that will be developed, there will be a similar diversity with regards to how they are made. One class of MECS devices with current appeal are portable and distributed heat-actuated thermal management systems. In particular, characteristics of the man-portable devices are that they be lightweight and made from materials with high thermal conductivity. Materials must be inert to working fluids and refrigerants such as LiBr, water and ammonia and generally operating conditions do not exceed several atmospheres and 250°C (other than combustors).

One promising avenue for addressing economical production in these types of devices is the application of surface mount technology (SMT). SMT is the practice and method of attaching leaded and non-leaded electrical components to the surface of conductive patterns in the electronic assembly industry. In the electronic industry, SMT enabled the trend toward electronic assemblies that have greater functionality, smaller size, less weight and less cost. In addition to being an efficient, economical platform for production, SMT also provides a platform for integrating electronics into MECS devices. This factor may become more critical as the need to integrate sensors and actuators within MECS devices grows. The bonding process in SMT requires a low temperature of about 300°C

and occurs at atmospheric pressure. The reflow process takes 2-3 minutes, which is negligible when compared to techniques like diffusion bonding which takes hours to bond laminae. The printing and reflow processes can be easily automated as well.

It is also proposed that low fabrication temperature and pressure, prevents warpage and residual stress in materials leading to more stable geometry and alignment. These devices when produced in high volume will have shorter production and throughput times. This will cut down the cost of the end product drastically. As shown in Figure 1, SMT can bring down the individual device cost by more than 50%.⁶ Figure 1 shows how the unit cost of MECS devices goes down when SMT is used for bonding as compared to diffusion bonding (DB) with photochemical machining (PCM) and blanking (BLK) as the patterning processes. This is mainly due to the decreased cycle time costs associated with the bonding process. The purpose of this paper is to see if SMT is viable for the production of microchannel devices.

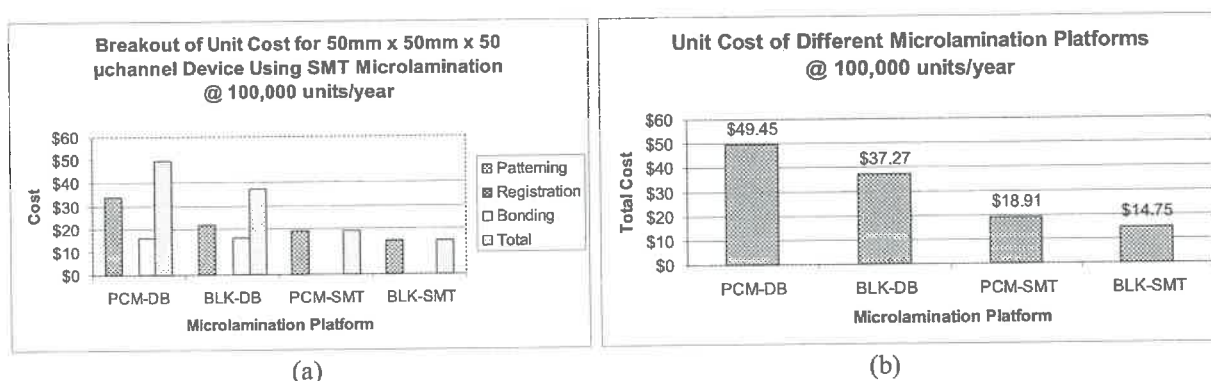


Figure 1: (a). Unit cost when separated into patterning, registration and bonding at a production rate of 100,000 units/year for four different platforms. (b). Unit cost for four different platforms.

MATERIALS AND METHODS

The substrate material chosen for this proof-of-concept experiment was Cu mainly due to the excellent wettability of Cu with standard solder paste. Cu shim stock approximately 200 µm (0.008 inch) in thickness was used for the experiments. The solder paste used for the tests is eutectic SnPb solder (Sn63-Pb37). Also, low residue no-clean flux is used for localized wetting of the Cu in some places where the solder paste is not applied.

The substrate obtained from the vendor was first cut into the required geometry using a Nd:YAG 355 nm laser mounted on an ESI 4420 laser micromachining system. To form a single microchannel, laminae with three different patterns were cut. The bottom plate was a flat Cu substrate with dimensions 4.4 x 1.9 cm. A Cu rectangular spacer with a rectangular window was used to form the channel. The outer dimensions of the spacer were 4.4 x 1.9 cm and the inner dimensions were 3.68 x 1.14 cm. The top inlet plate had dimensions of 4.4 x 1.9 cm with a small diameter hole of 0.18 cm at the center for leakage testing. All of these laminae were deburred using Scotch Brite after being cut on the laser. These deburred laminae were then flattened in the hot press at a temperature of 500°C and a pressure of 36.75 bars.

Solder paste (Sn63-Pb37) was printed on each lamina using a Semi-Automatic Ekra screen printer. Low residue no-clean flux was applied at the tie-bar areas where the solder paste is required after reflow. These solder printed laminae were stacked on the top of each other in the desired pattern and then reflowed through the Quad ZCR convection based four zone reflow oven. The device was pressure tested for leakage at 1.72 bar with the help of a fixture.

RESULTS

The flattening procedure on the hot press for the Cu substrate showed significant improvement in the flatness and roughness of the laminae. This was critical for proper solder paste printing. The fresh shim stock received from the vendor had a mean flatness of 99.9 ± 32.9 µm. After treating the laminae on the hot press, the flatness obtained had a mean of 4.73 ± 1.14 µm. Figure 2 shows the flatness profiles obtained before and after treatment with the hot

press. Further, as shown in Figure 3, the burrs produced by the laser were greatly reduced by using the mechanical deburring technique going from a mean of $19.1 \pm 8.5 \mu\text{m}$ to $2.93 \pm 2.3 \mu\text{m}$.

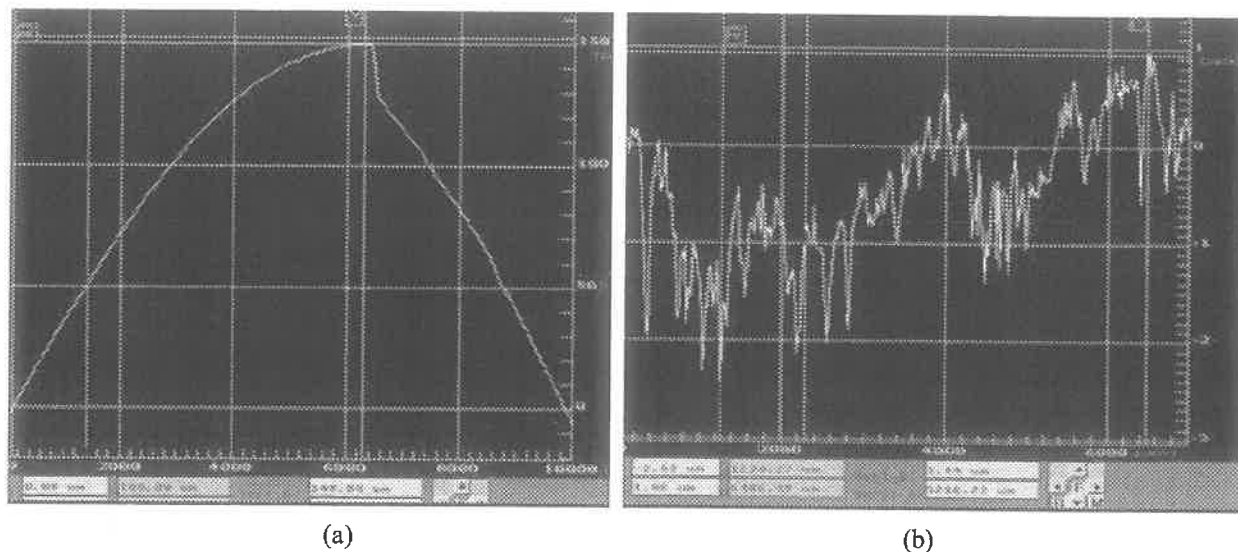


Figure 2: (a). Flatness variation of a fresh Cu shim obtained from vendor. (b). Flatness variation of a treated Cu shim.

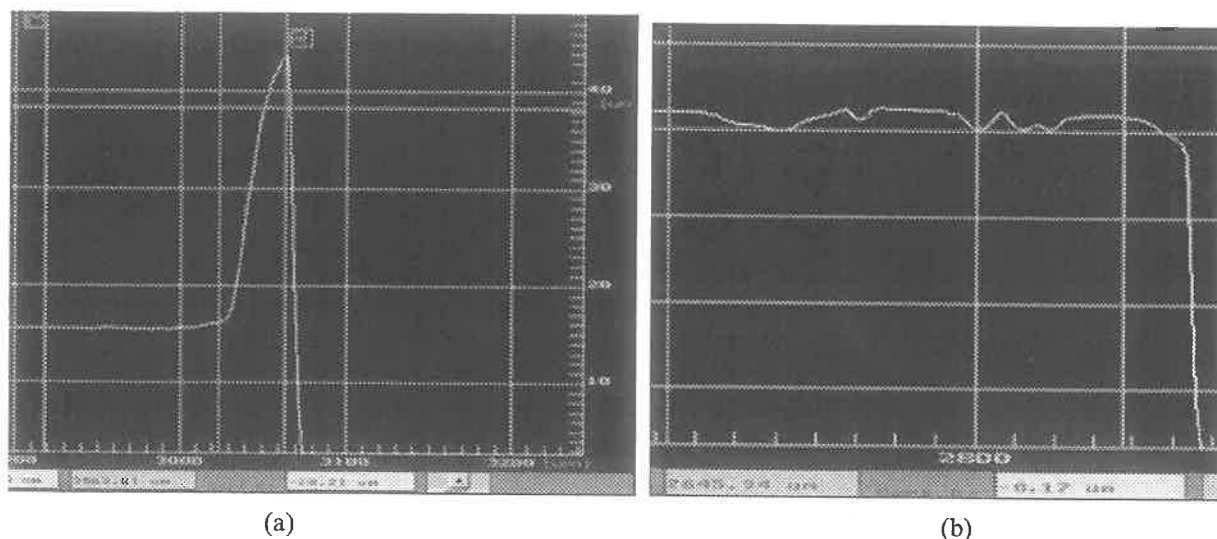


Figure 3: (a) Profile obtained from laser burr on the Cu shim. (b) Profile obtained after deburring the Cu shim.

Figure 4 shows the single microchannel made by application of SMT. A channel height of $334.3 \pm 3.14 \mu\text{m}$ was obtained using this process. This indicates a channel uniformity of less than one percent which is currently unattainable in diffusion bonding based processes. Also the device successfully passed the leakage test when tested at a pressure of 1.72 bars. More recently a leak proof (pressure tested at 1.72 bar) four-channel microchannel device was also made using the same technique.

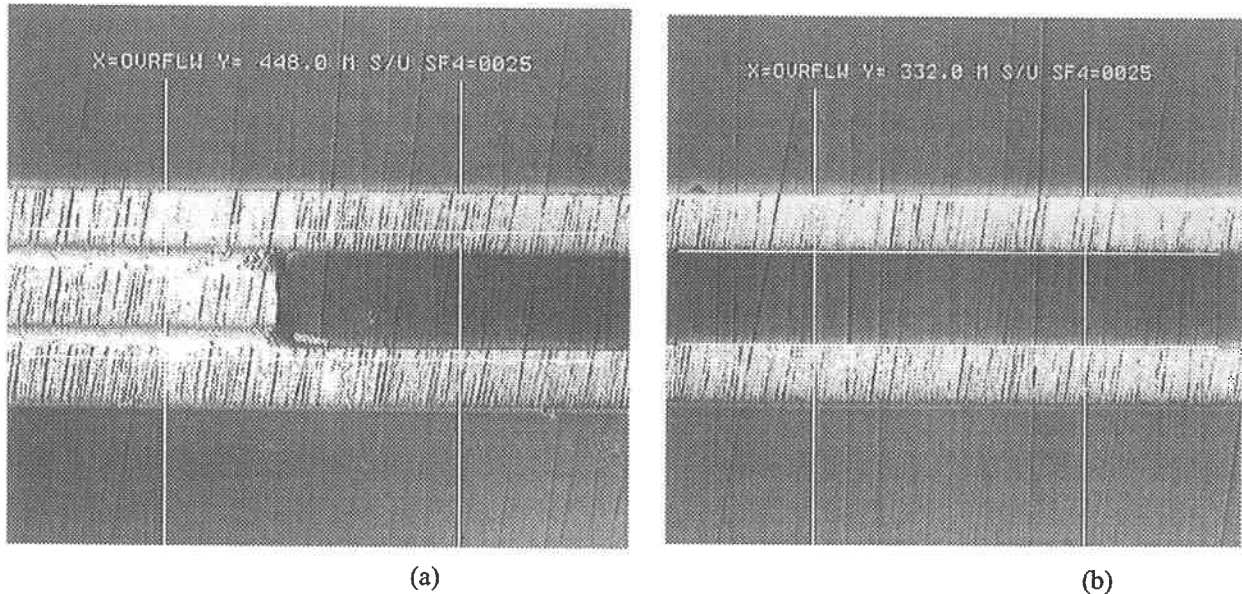


Figure 4: (a) The spacer with the solder layer and the channel. Solder joints are approximately 65 μm in height. (b) Picture of the midsection of the channel showing excellent uniformity. Overall channel height was 332 μm with a channel aspect ratio of 34:1.

CONCLUSIONS

The results obtained show that SMT can be successfully used to produce high aspect ratio microchannels. The use of low bonding temperatures (in the range of 300°C) and pressures results in well aligned, parallel channels with extremely low levels of warpage. Also, the surface flattening procedure using the hot process, resulted in significant reduction in the surface roughness of the laminae. Economic analyses indicate that this process will be favorable over existing diffusion bonding-based processes. While this material and process combination will not meet all of the requirements for future MECS devices, it has shown the potential to use solder pastes in the formation of microchannels. Future efforts will be focused on understanding the physics involved with controlling microchannel height in multi-channel arrays as well as the development of new solder paste/substrate material combinations acceptable for manportable MECS applications.

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A Comparison of the Mass Transfer Performance of Conventional and Micromanufactured Porous Membranes

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ABSTRACT

Recent findings have suggested that the performance of absorption/desorption cycle micro-scale heat pumps can be significantly enhanced by the use of thin film gas/liquid contactors. In this study, it was hypothesized that microfabricated straight-through pores could significantly reduce the pressure drop for gas diffusion across a contactor as compared to the complicated and tortuous flow paths encountered in commercially available membranes. Below a comparison is made between the mass transfer performance of conventional membranes with those having engineered pores in an attempt to classify the mass flux characteristics of contactor membranes by the membrane morphology. Results show that the mass flux properties of woven membranes far exceed those of even engineered membranes.

Keywords: membranes, tortuosity, mass flux characteristics, contactors, absorption, desorption, heat pumps

INTRODUCTION

Microtechnology-based Energy and Chemical Systems (MECS) are massively-parallel microsystems fabricated in engineering materials capable of processing bulk amounts of fluid in microchannels.^{1,2} MECS technology enables process intensification (i.e. the shrinking of energy and chemical process footprints) in an effort to distribute, decentralize and make portable energy and chemical systems such as heat pumps, chemical plants, and cytosensors. The miniaturization of MECS devices is made possible by the manifold improvement in the heat and mass transfer performance of systems due to the high surface area-to-volume ratios of the microchannel arrays within the devices.

Recent findings have suggested that the performance of absorption/desorption cycle micro-scale heat pumps can be significantly enhanced by the use of thin film gas/liquid contactors. However, a key limiting factor in the use of thin film gas/liquid contactors is the driving force across the contactor membrane with for the heat pump systems of interest is quite low (a few psi). Therefore, there is currently a great interest in understanding the nature of mass flux characteristics across contactor membranes. One hypothesis in this work was that a contactor membrane microfabricated with straight-through pores could significantly reduce the pressure drop for gas diffusion across a contactor as compared to the complicated, tortuous flow paths encountered in commercially available membranes. Below, two approaches to membrane fabrication are explored along with a comparison between the mass transfer performance of conventional and engineered membranes. The morphology of each membrane was studied by micrograph and related to the tortuosity factor calculated from mass flux measurements. The two engineered membranes are fabricated using laser micromaching and micromolding. The final outcome is an attempt to classify the mass flux characteristics of contactor membranes by membrane morphology.

EXPERIMENTAL APPROACH

A baseline for mass flux measurements was established by testing twenty-three different conventional membranes. To parse the enormous list of commercial membranes, some selection criteria were set that would most approximate the mass flux conditions for gas/liquid contacting. The selection criteria included that they be made from polymers with thicknesses between 50 and 150 μm and pore sizes from 1 to 10 μm .¹ Both had been tested and compared to the baseline of the mass flux measurements. No consideration was given up front for the chemical hydrophobicity (or hydrophilicity) of the membrane. In this study, only pressure drop measurements of membrane morphology were studied.

A test loop was developed capable of measuring the pressure drop across a membrane as a function of mass flux. The experimental setup for conducting the pressure drop testing across the conventional membranes involved flowing nitrogen from a tank through a needle valve (flow rate control) and into the lower plenum of a test fixture. Once in the test fixture, the nitrogen flowed across the membrane and into the upper plenum of the test fixture. The test fixture was necessary to ensure flow across a constant cross-section of each membrane. The diameter of the upper plenum was 2.0 inches while the lower plenum had a diameter of 0.4 inches. A stiffener was added to prevent the membranes from deflecting. The stiffener was 500 microns thick made from partially sintered stainless steel powder with a final pore size of 100 micron. An OMEGA HHP-2000 digital manometer was used to measure pressure drop between the upper and lower plenums. A

MKS Type M10MB (M10MB13CS3BV) mass flow meter was connected to the test loop at the output of the upper plenum. The test loop was designed to permit no more than 5% uncertainty in pressure drop (between 2 and 18 torr) and mass flux (between 200 and 1000 sccm).

Once a membrane was secured within the test fixture, the testing protocol involved first running the experiment up to the maximum recordable pressure drop or flow rate and then back down to zero. Between five and ten data points were collected in both directions.

MEMBRANE FABRICATION

Laser Micromachining Approach – A critical parameter for consistently producing 5-15 micron holes across the surface of the membrane is the depth of field (Z_R). The Gaussian beam expands rapidly beyond two times of the depth of field region. Assuming a 266 nm laser, a 0.5-inch focal length lens and an incident beam diameter of 1 mm, one half of the depth of field is found to be 53 micron. Therefore, theoretically, the total depth of field under laboratory conditions is only 106 μm . This is acceptable since the depth of the material used within this study was significantly less than 100 μm . Also, in order to consistently machine the material, a procedure was developed for consistently focusing the laser beam onto the workpiece surface. In addition, efforts were made to reduce the flatness of the workpiece material to within 100 μm .

Micromolded Approach – The PDMS membrane fabrication process can be divided into three major steps. The first step is to create SU8 posts on the substrate. Second step is to spin coat PDMS on to the substrate with SU8 posts in order to create PDMS membranes with straight through pores. The third step is to separate PDMS membrane from the SU8 mold and substrate. The permeability was evaluated from the test loop.

DATA NORMALIZATION

Conventional membranes tend to have tortuous flow paths and so it is expected that they will have a higher pressure drop than membranes with straight-through pores. However, other factors can affect the pressure drop across a membrane. For example, it is expected that thicker membranes will have higher pressure drops. In addition, membranes with higher pore packing density are expected to have lower pressure drops. In order to objectively analyze the pressure drop data collected, several equations were developed to help normalize the data.

The first consideration in normalizing the data is to consider the impact of open area or pore packing density. This parameter is easy to quantify for straight-through pores as it is the pore cross-sectional area times the number of pores per unit membrane area. For fibrous or other tortuous path pores, it is more difficult to quantify. In this thesis, open area or pore packing density is quantified by the fractional density of the membrane. The fractional density of the membrane is equal to the measured density of the membrane divided by the density of the membrane material.

The effect of the fractional density is to impact the total mass flux as follows:

$$\text{mass flux} = \frac{\text{mass flow}}{A_o} = \frac{\text{mass flow}}{(1-f)A} \quad (2)$$

where A_o is the actual open area through which the fluid flows, f is the fractional density of the membrane, and A is the nominal area of the membrane exposed to the fluid.

The effect of the membrane thickness is more straight forward. Assuming laminar flow conditions, pressure drop through a pore is assumed to be proportional to:

$$\Delta p \propto \frac{h \cdot V^2}{D_H} \quad (3)$$

where D_H is the hydraulic diameter and V is the average velocity through the channel. To compare membranes, we normalize pressure drop by dividing through by the membrane thickness, h .^{3,4,5}

$$\frac{\Delta p}{h} \propto \frac{V^2}{D_H} \quad (4)$$

These two normalizing procedures are used to help identify patterns in the data and to draw conclusions about the effect of membrane morphology on the mass flux characteristics of the membrane.⁵

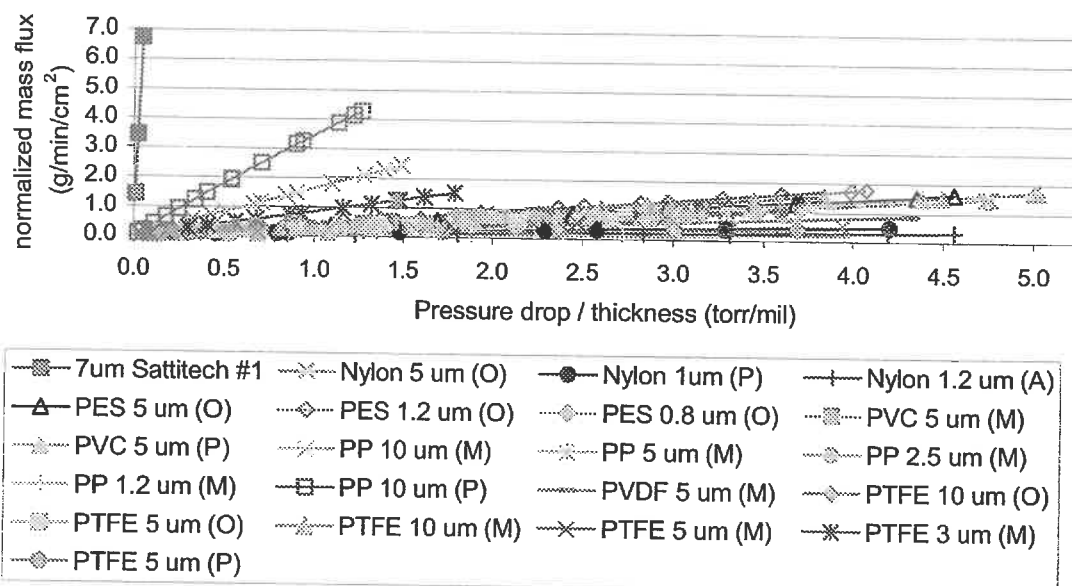


Figure 1. Normalized pressure drop results across the baseline membranes.

RESULTS AND DISCUSSION

Baseline Measurements – Figure 1, shows the normalized pressure drop results across each membrane using equations 2 through 4 above. An analysis of the morphology of these membranes showed that the membrane morphologies can be classified into three broad categories: woven, fibrous and networked. Networked membranes differ from fibrous membranes in that the membrane is an interconnected web of material while fibrous membranes are not connected with one another. One difference in these morphologies is that networked morphologies are sponge-like and may result in “dead ends” or flow terminations as the fluid winds through the structure. Fibrous morphologies are incapable of terminating flow paths. Therefore, it is plausible that the networked structures could provide higher pressure drops than fibrous structures.

Figure 2 shows the same normalized graph plotted for the three different morphologies investigated in the mass flux baseline. It supports the notion that in general, fibrous morphologies provide less pressure drop than networked morphologies. All of the better performing membranes are either woven or fibrous with woven membranes significantly outperforming the fibrous membranes.

Laser Micromachining Approach – A nominally 75 μm thick Kapton membrane was laser micromachined with an array of 22,500 straight-through holes (150 by 150 holes). The average pore size was found to be different on the front side than on the back side of the material due to the characteristic parabolic shape of the laser energy deposition. Using optical microscopy, the average pore size was found to be $16.5 \pm 0.5 \mu\text{m}$ for the front side and 5.3 ± 0.8 for the back side at a 95% confidence interval (See Figure 3a and b).

In general, the mass flux results of the laser micromachined membrane did not show a significant

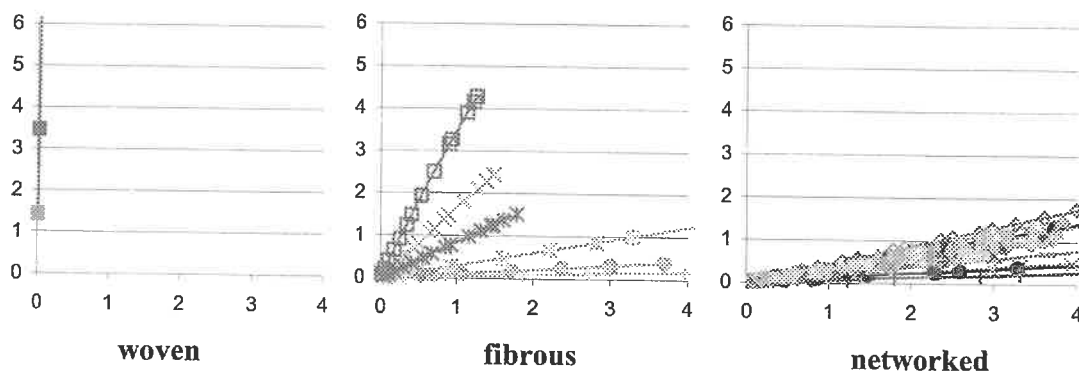


Figure 2. Normalized pressure drop results for (from left to right) woven, fibrous and networked structures

improvement over the commercial woven and fibrous membranes. In comparison to the networked membranes with similar pore sizes, the pressure drop was found to decrease by several times. It was expected that the straight through pores would have superior performance as compared to the fibrous and networked membranes. Several explanations for its unexpected performance are given here. First, since the laser ablation process is a thermal process, it is expected (and can be verified with microscopy) that the surface roughness on the inside of the hole is greater than the surface roughness of either the woven or the fibrous membranes. This observation might provide additional insight as to why the fibrous membranes perform better than the networked membranes. At small pore sizes, the pressure drop effects of surface roughness are magnified if the flow is turbulent. A second explanation comes from the observation that some variation in pore size might exist across the membrane. While the initial results of pore size across the membrane are encouraging, it is impossible to check all 22,500 pores on the membrane. Prior tests have shown that if the membrane is not within its required flatness, the out-of-focus can cause large variations in pore size including elimination of the pores altogether. Additional methods are being investigated to minimize workpiece flatness prior to laser processing.

Micromolded Approach – Silicon wafer substrates were chosen to spin coat SU8. A photomask of hole arrays consisting of 5-micron diameter holes on 100-micron spacing was used to expose the resist to create a SU-8 mold (Figure 3c shows a 45 degree view of the posts). PDMS is then spincoated into the SU8 micromold (see Figure 3d) and then the PDMS membrane is released.

Several membranes were fabricated with aspect ratios as high as 3:1. However, no flow was reported in any of the membranes as the minimization of energy in the membranes with 5 micron holes (in nitrogen and air) caused the holes to collapse. In the future, stiffer PDMS formulations will be considered to avoid the sealing of the pores to themselves.

CONCLUSIONS

Based on test results to date, we have found that, in general, woven pore membranes have much higher mass transfer rates than all other membranes. We have classified commercial membranes into three key classes: woven, fibrous and networked. We have developed an initial normalization procedure for comparing the results of various tests. Also, we have found that the laser micromachining technique improves the mass transfer rate in general but that the mass transfer properties are much less than those found for woven membranes. Conventional PDMS technology was not found to support membranes with such high aspect ratio. Future work could involve the formulation of membranes with stiffer PDMS properties.

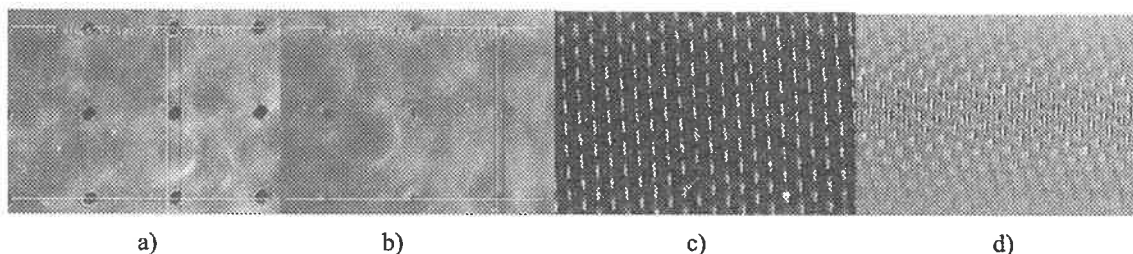


Figure 3. a) Front of pore array with 100 μm spacing and average pore size of 16.5 ± 0.5 ; b) Back of pore array with 100 μm spacing and average pore size of 5.3 ± 0.8 ; c) Micrograph of 5 μm diameter posts with a height of 65 μm (1:13 aspect ratio) on a silicon wafer; and d) PDMS spincoated onto SU8 micromold with a height of approximately 35 μm .

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NOVEL ULTRAPRECISE TOOL ALIGNMENT SETUP FOR CONTOUR BORING AND BALL-END MILLING

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Abstract

Diamond turning and fly-cutting extend the spectrum of machinable surfaces and are commonly employed for ultraprecision manufacturing of complex surfaces and microstructures [1,2].

In contour boring and ball-end milling the achievable figure accuracy strongly depends on the geometry of the cutting edge and on tool alignment. Up to now, tool alignment is performed with available standard precision components. However, when large tool radii and/or high rotational speeds are employed, the limited stiffness and imbalance of tool holder designs based on such components eventually leads to chatter and process instability [3].

In this paper the development of a novel ultraprecision tool alignment setup is shown and first machining results are presented.

Introduction

Contour boring is performed by infeeding a rotating half-arc monocrystalline diamond tool along the axis of rotation. The contour of the cutting edge is directly copied into the surface. Ball-end milling is realized by superimposing the rotational motion with a lateral feed motion (fig 1.).

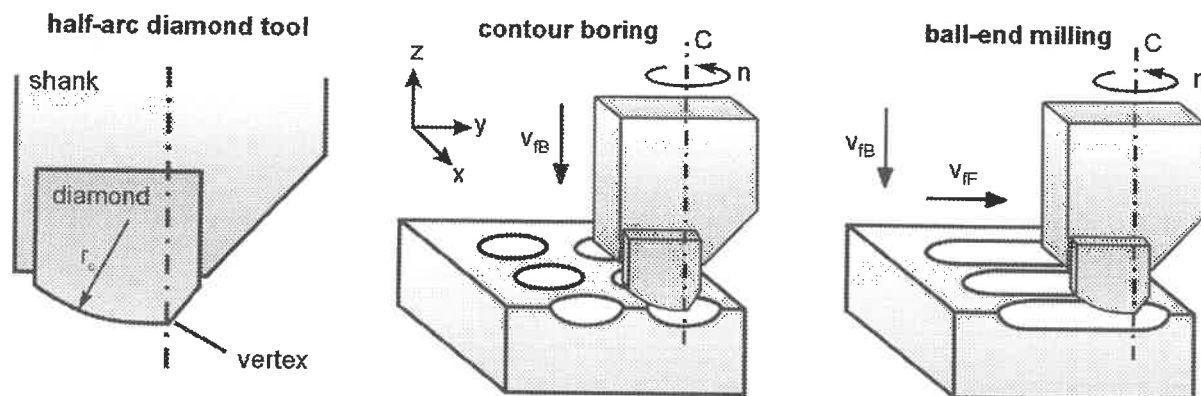


Fig. 1: Contour boring and ball-end milling with half-arc monocrystalline diamond tools.

The achievable figure accuracy depends on the geometry of the cutting edge and on tool alignment. The vertex of the cutting edge must precisely be aligned to coincide with the spindle axis. Moreover, the angular orientation of the diamond tool with respect to the spindle axis has to be aligned in order to avoid ogive shapes of the machined cavity. Fig. 2 shows this type of misalignment and the resulting contour errors.

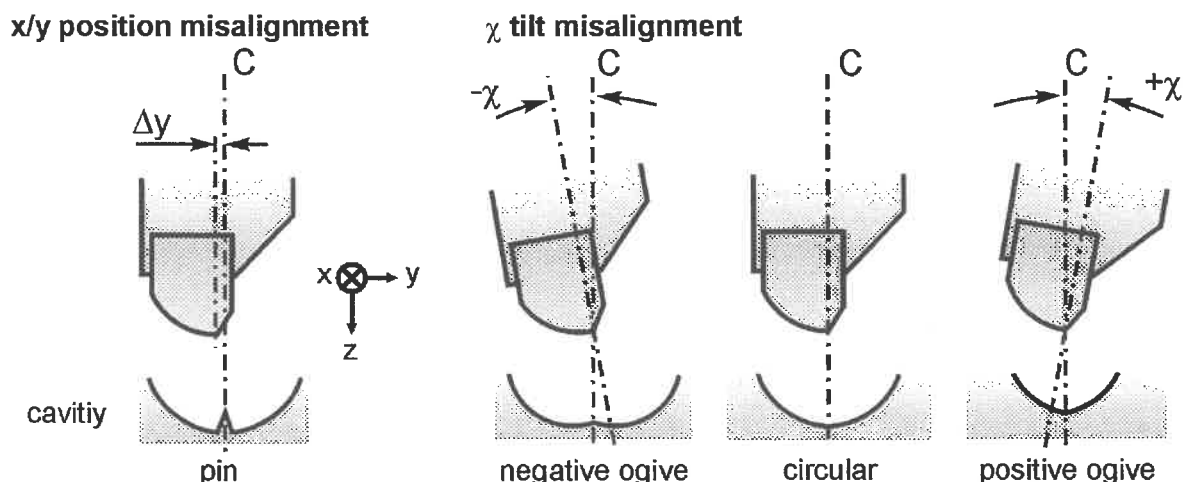


Fig. 2: Position and tilt misalignment.

First tool alignment experiments were performed with a standard precision linear-slide, a rotary table and a spherical bearing for tilt adjustment (fig. 3). The position of the tool vertex was measured with an optical microscope and was aligned by moving the vertex towards the spindle axis by means of the linear slide and rotary table ($\Delta\phi$, $\Delta x'$). Tilt alignment was done by measuring the shape of bored cavities with a white-light interferometer (WLI), calculating the tilt misalignment χ and correcting the tilt correspondingly. In successive steps of centring, boring and tilt correction a figure accuracy $PV \approx 0.2 \mu\text{m}$ could be achieved.

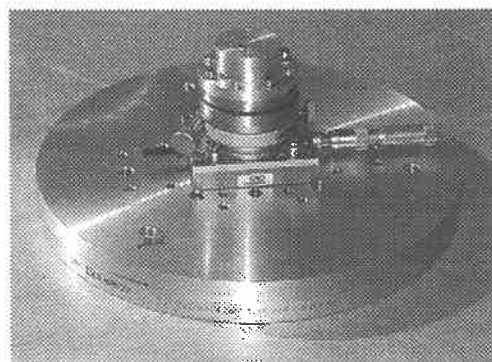
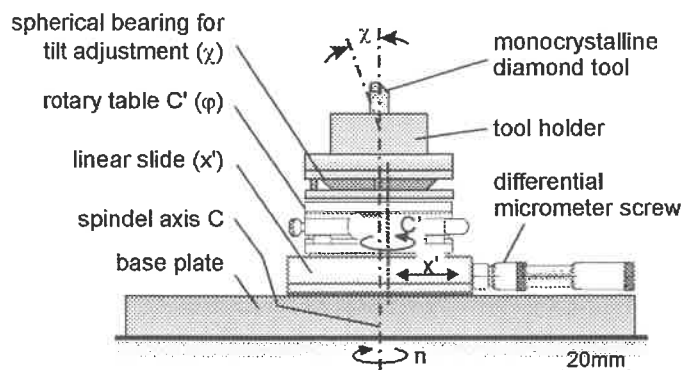


Fig. 3: Adjustable tool holder composed of standard precision components.

Unfortunately, large tool radii, hard workpiece materials and high rotational speeds lead to figure deviations and chatter marks. This is caused by high thrust forces compared to the relatively low stiffness of the alignment setup.

Novel Tool Alignment Setup

For overcoming imbalance and stiffness limitations, a novel ultraprecision tool alignment setup was developed. The new setup consists of two moveable ring elements which are arranged in different off-axis positions, such that moving one or the other of these elements does not affect the overall balancing state of the setup. The half-arc diamond tool is clamped in the tilt adjusting device placed in the inner ring element of the setup

(fig. 4). By rotating the respective ring element the tool can be moved in the x/y plane within an area of a few square-millimeters with respect to the rotational axis C with a positional accuracy $< 0.2 \mu\text{m}$. The movement of each ring element is realized by a pinion gear and a gear rim. High stiffness of the system is guaranteed by using conical ring elements with a 15° cone angle for self-locking. The tilt is adjustable in two degrees of freedom by a spherical guiding device with four differential micrometer screws. Tilt errors can be corrected within a range of $\pm 3^\circ$ with an accuracy $< 0.05^\circ$. The stiffness is $\approx 25 \text{ N}/\mu\text{m}$, compared to $\approx 5 \text{ N}/\mu\text{m}$ for the conventional system. The tool alignment procedure was done in the same way as described above. The balancing state of the setup was monitored with an accelerometer.

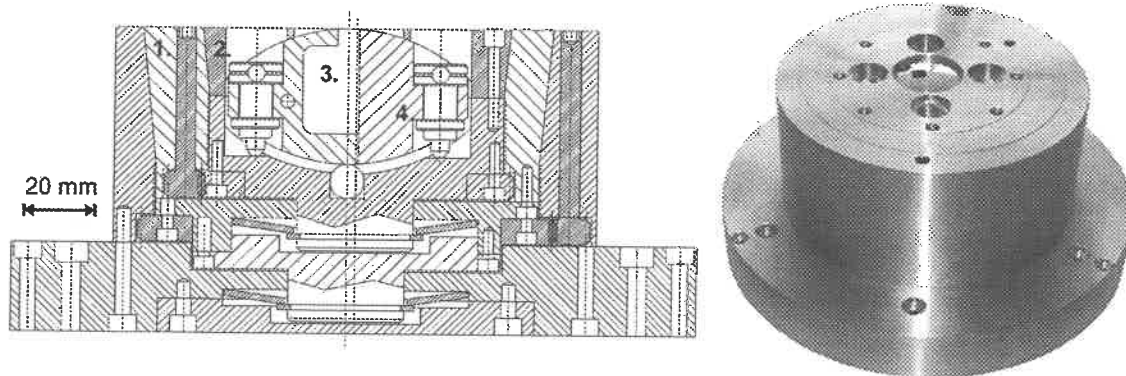


Fig. 4: Novel adjustable tool holder (1. outer ring element; 2. inner ring element; 3. tool holder; 4. device for tilt adjustment).

Results

Contour boring and ball-end milling with the novel setup is shown in fig. 5. Spherical cavities were contour bored in an electroless nickel plated steel specimen with half-arc monocrystalline diamond tools with an nose radius $r_n = 80 \mu\text{m}$. Feed velocities between $v_{fB} = 0.02 \text{ mm}/\text{min}$ and $0.5 \text{ mm}/\text{min}$ and rotational speeds up to $n = 2000 \text{ min}^{-1}$ were employed. For ball end milling a horizontal feed between $v_{fF} = 1.0 \text{ mm}/\text{min}$ and $2.5 \text{ mm}/\text{min}$ was applied. The achieved roughness was $R_a = 4 \text{ nm}$ measured along the flank of the cavity over an area of $0.98 \text{ mm} \times 1.29 \text{ mm}$ with an WLI.

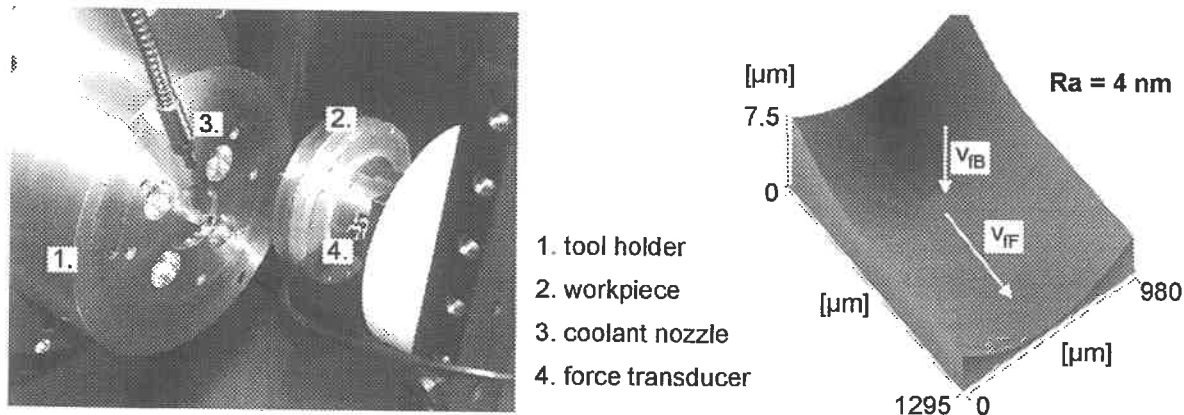


Fig. 5: Contour boring and ball-end milling (left: machining setup, right: WLI-image of milled cavity) with the novel tool alignment setup.

Thrust forces were measured with a high resolution force transducer with an operating threshold < 0.01 N. The thrust forces were found to be less than 3 N, essentially free from superimposed vibrations.

Cavities bored with the conventional and the new setup are shown in fig. 6. The figure accuracy obtained with the novel design was $PV \approx 100$ nm compared to $PV \approx 200$ nm with the conventional setup.

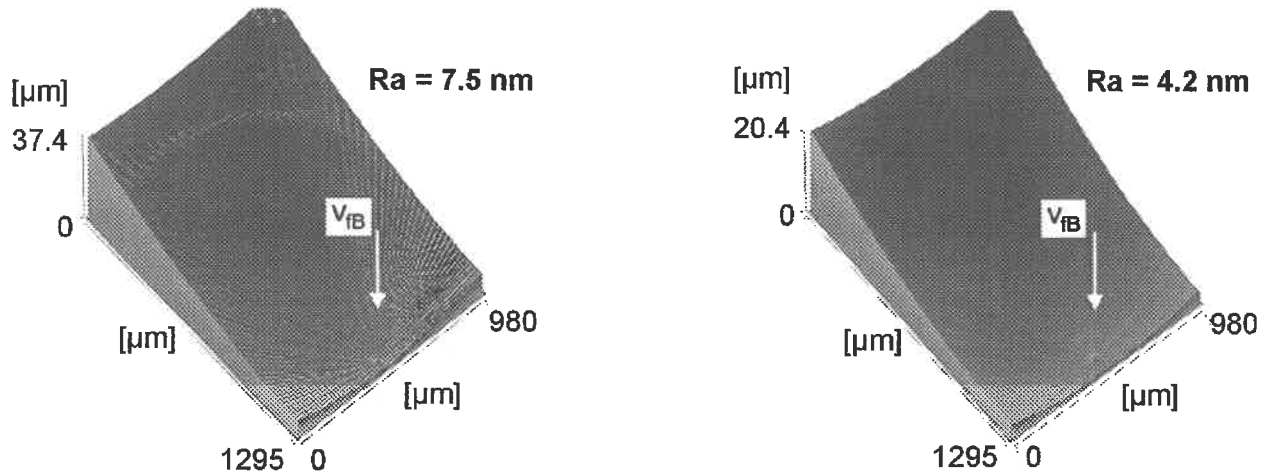


Fig. 6: Bored cavities in electroless nickel with identical machining parameters (left: conventional setup, right: novel setup).

Conclusion

The paper shows the design of a novel tool alignment setup for contour boring and ball-end milling and describes the alignment procedure. The vertexes of half-arc diamond tools can be positioned with an accuracy < 0.2 μm in the x/y plane. The angular orientation is adjustable with an accuracy $< 0.05^\circ$. The advantages of the new setup are higher stiffness and excellence balancing independent of the actual tool position. Therefore, high rotational speeds up to 2000 min^{-1} can be applied in contour boring and ball-end milling with tool nose radii up to 80 mm.

Acknowledgement

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LARGE DISPLACEMENT FLEXURE BASED NANO-PRECISION MOTION STAGE FOR VACUUM ENVIRONMENTS

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Abstract

This paper presents a large displacement flexure based high precision motion stage for vacuum-based semiconductor equipment. The weight support mechanism of the motion stage is made of links and flexure joints, and a linear motor is used as the actuator. This stage is not a dual servo stage, but it can provide large motion range of greater than 200mm to support a substrate during semiconductor manufacturing. An over-constrained mechanism is used to incorporate symmetry to cancel out the effects of center shifting in large motion flexures. Experiments are performed to characterize motion straightness and repeatability of the motion stage actuated by a high-resolution linear motor with a laser interferometer providing real-time position feedback.

1. INTRODUCTION

Lead screws and ball screws have been traditionally used in motion stages, but both have two major drawbacks, backlash and stick-slip friction that decrease motion accuracy. Further, these systems undergo wear over time, and lubrication makes it undesirable for vacuum or particle-sensitive applications. Friction drive systems that cause wear and generate particles at the contact surfaces are not good in vacuum environments. Since air bearings do not have backlash and stick-slip friction, most of the commercial high precision motion stage suppliers now use some form of air bearing. However, it is difficult and expensive to use air bearings inside vacuum environments, because it requires complicated design to take away exhausted air. A new approach using magnetic levitation has been recently proposed to enhance the position accuracy of motion stage and to eliminate undesirable dynamic effects such as friction and backlash. This approach has physical limits such as thermal dissipation problems and inherent interference with electron beams in commercial electron beam based patterning and inspection equipment.

Since linear motors are completely non-contact devices, there is no friction, no cogging, and no parts to wear. As a linear-motor-based system can provide high speeds and accelerations, linear motors are becoming the best actuators for ultra-high precision applications. Flexures are widely used in precision machines since they offer frictionless, particle-free and low maintenance operation, and they provide extremely high resolution. Since notch type flexure joints are ideal for small motion range, all positioning stages developed so far are useful for a few hundreds of micrometer range. To achieve high precision in large motion range, many researchers have studied dual servomechanism that has a fine motion stage mounted on a coarse motion stage. They used flexure joints for guidance in fine motion stages. Obviously dual servo stage makes the whole stage complex, and errors associated with coarse motion stage cause undesirable effect on the whole stage.

The Free-Flex Pivot (see Figure 1) made by Lucas Aerospace [2] is an example of a commercially available large deformation revolute flexure joint. This flexure does not have friction or backlash, and provides a large rotation of 60°. These flexure joints do not provide exact rotary motion over the entire 60° motion range. They can, however, be used in conjunction with symmetric kinematic designs to yield exact linear motion stages.

Large displacement flexure based nano-precision motion stage for vacuum-based semiconductor equipment is developed in this research. The weight support mechanism of this motion stage is made of links and flexure joints, and a linear motor is used as the actuator. This stage is not a dual servo stage, but it can provide large motion range of greater than 200mm to support a substrate during semiconductor manufacturing. Novel kinematic synthesis techniques have been used to incorporate symmetry to cancel out the effects of axis shifting in large motion flexures. Advanced kinematic techniques such as screw system theory have been used to achieve a good kinematic design. Experiments are performed to characterize motion straightness and repeatability of the motion stage actuated by a high-resolution linear motor with a laser interferometer providing real-time position feedback.

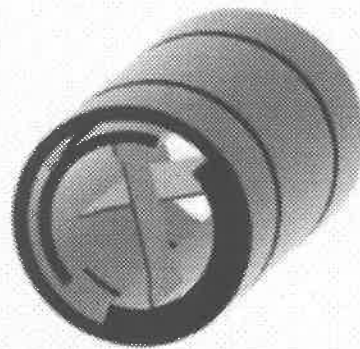


Fig. 1. Crossed strip type flexure joint

2. DESIGN OF FLEXURE BASED MOTION STAGE

Figure 2 shows a double compound notch type small motion rectilinear spring that is selected as the basic design configuration for the new motion stage. Since semi-circular notch type flexures are ideal only for small motion range, large-motion flexure joints such as the one shown in Figure 1 are used here.

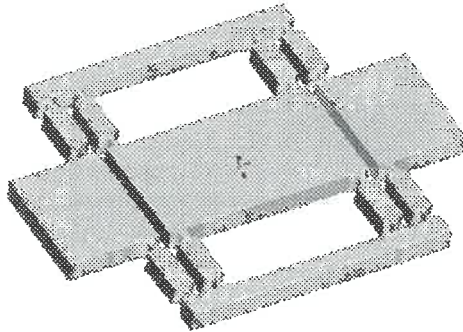


Fig. 2. Double compound notch type small motion rectilinear spring

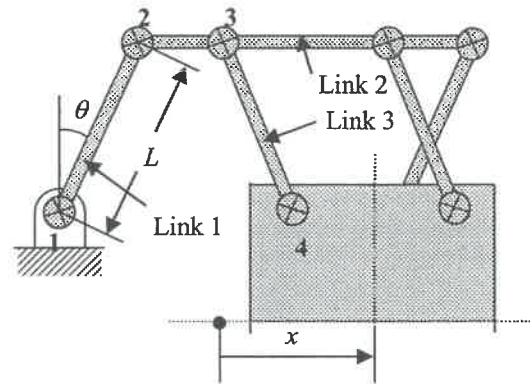


Fig. 3. Linkage schematic for the stage

Assuming that constraining linkages exist, the mechanism in Figure 2 can be optimized to lead to the smallest footprint for a 300mm motion range. Figure 3 shows a schematic of the basic linkage in Figure 2. The links 1 and 3 are of the same length. When the moving body moves along the X axis, all the joints shown in Figure 3 rotate by the same absolute angle. It should be noted that the motion range is independent of the length of the link 2. Due to kinematic constraints, the link 2 remains parallel to the line connecting the joints 1 and 4. Therefore, the link length is written by

$$L = \frac{x_m}{\sin(\theta)} \quad (1)$$

where x_m is the range of motion, θ is the angle of the joint 1, and L is the length of link 1. The minimum link length for a 300mm motion range is about 150mm when $\text{abs}(\theta_{\max})$ is 30° .

The moving body has one degree-of-freedom in its nominal configuration and has been used for small motion applications [4]. A mobility analysis based on Screw System Theory [1] shows that the moving body has two degree-of-freedom in its off-nominal configurations. Therefore, the undesirable degree-of-freedom must be eliminated for a large motion application. It is necessary to use additional linkages to eliminate the undesirable motion and constrain the moving body along the linear motion direction. Therefore, side

links are installed at both sides of the moving body. The "Side Links" shown in Figure 4 eliminate this undesirable degree of freedom, which makes this stage one degree of freedom. These side linkages are oriented orthogonal to the XY plane and move in the XZ plane.

Incorporation of symmetry can lead to undesirable singularities in mechanisms [1]. From preliminary assembly of the proposed motion stage, an unexpected independent mode vibration problem occurred. As shown in Figure 5, the link 2 can vibrate independently in the nominal position. In order to prevent this independent vibration mode, the flexure joints are installed in the off-nominal configuration. This makes the nominal position outside the motion range. Total travel range was reduced to 200mm because of interference between linkages. Figure 6 shows the solid model of the resulting motion stage design.

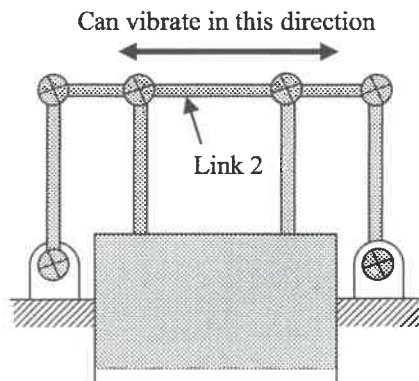


Fig. 5. Independent vibration mode in nominal position

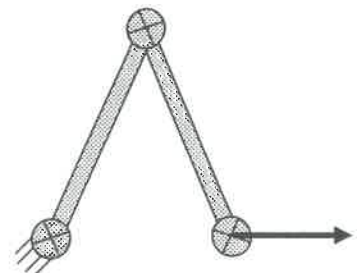


Fig. 4. Schematic of side linkage

3. EXPERIMENTS

An ideal single-axis motion stage should give only motion in the X axis, but an actual linear motion stage has three translational and three rotational errors, such as X axis servo error (δ_x), horizontal straightness error (δ_y), vertical straightness (δ_z) error, roll (θ_x), yaw (θ_z), and pitch (θ_y) [3]. Since the proposed stage is symmetric about the X axis, roll is less significant than yaw and pitch. The horizontal straightness, yaw, and their repeatability are observed experimentally by using the laser interferometer. If these undesirable motions are repeatable, a lookup table can eliminate such errors. If not, it is necessary to do a complicated coordinated control of the two stages which is more difficult to be implemented in real-time at high bandwidths.

A single-axis motion stage is fabricated to verify the control performance of the proposed flexure stage experimentally. The BLM-203-A linear motor manufactured by Aerotech Inc. is used as an actuator to move the stage. The BA20-160 amplifier produced by the same company generates input power into the linear motor. The Agilent 10889B servo axis board is employed as the motion controller to send the control signal to the amplifier. A digital PID control algorithm is employed to control the position of the stage.

Figure 7 illustrates a photograph of the experimental setup for the horizontal straightness, yaw, and repeatability test. As the stage is moved step-by-step at the increment of 10 mm for a distance of 100 mm, for 3 runs, the lateral displacements (Y1 and Y2) at each step are measured using the laser interferometer. Figure 8 shows the straightness error test result (Y1). Figure 9 shows the measured values of the yaw of the stage. The straightness error and yaw test results show very repeatable behavior.

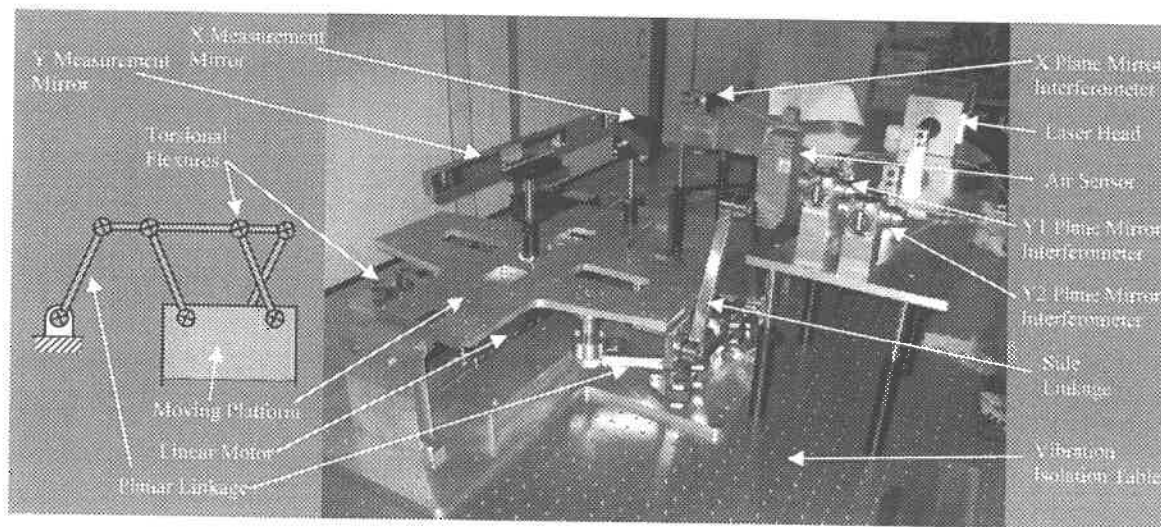


Fig. 7. Photograph of experimental setup

The repeatability of the horizontal straightness and yaw of the stage is also examined. Moving the stage forward and backward twenty times, the Y axis displacements at 50 mm from the home position are measured. Figure 10 shows the repeatability test results of the horizontal straightness and yaw. The standard deviation (σ) is about 1.29 micron for Y1 and 0.0015 milli-radian for yaw. The repeatability (3σ) of the horizontal straightness of the stage is about 3.9 micron, but the stage shows a very excellent yaw repeatability of 0.0045 milli-radian. The

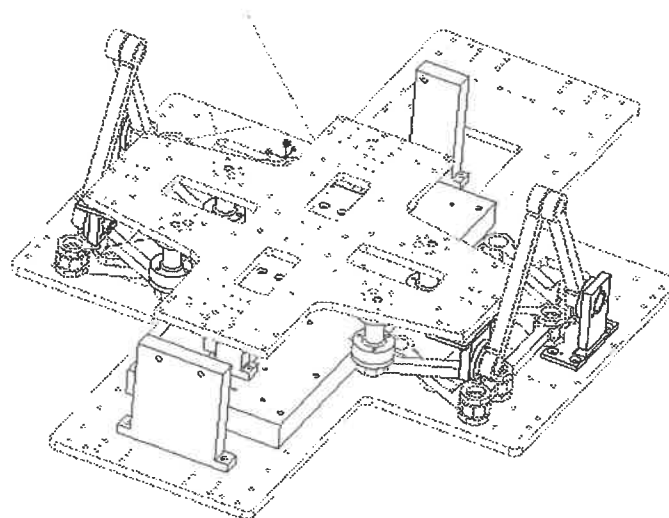


Fig. 6. Solid model of modified motion stage

major reason for relatively large straightness repeatability error is low lateral stiffness of the current flexure stage. Therefore, the design should be modified to increase this lateral stiffness.

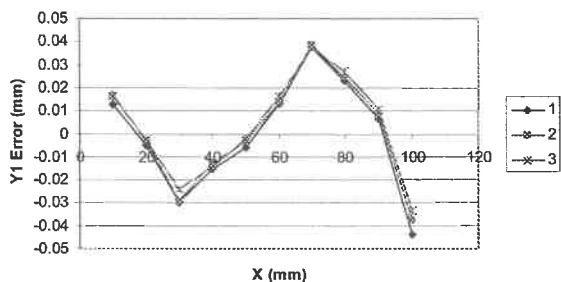


Fig. 8. Straightness error test result (Y1)

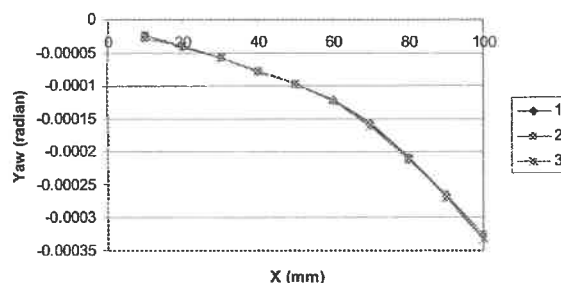


Fig. 9. Yaw test result

4. CONCLUSIONS

A large displacement flexure-based motion stage has been developed for precision equipments in vacuum environments. This research is the first work for developing a macro motion stage that can support the weight of the stage and guide the motion by a mechanism based on flexure joints. A symmetric over-constrained mechanism is used to eliminate center shift in flexure based system. From experiments, it is shown that the proposed motion stage can be an excellent substitute for a precision motion stage especially in vacuum environments. The horizontal straightness, yaw, and their repeatability tests are performed. Since the low lateral stiffness of the current stage causes relatively large straightness repeatability error, it is necessary to modify the design to increase the lateral stiffness.

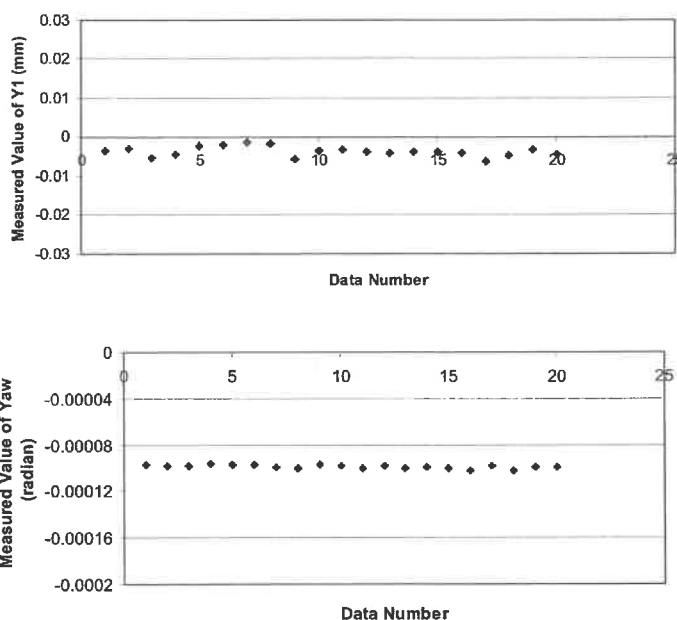


Fig. 10. Repeatability test result

ACKNOWLEDGEMENT

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DESIGN OF PRECISION MANIPULATORS USING BINARY ACTUATION AND DIFFERENTIAL COMPLIANT MECHANISMS

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Keywords: Compliant mechanism, Digital actuation, Repeatability, Accuracy

1. INTRODUCTION

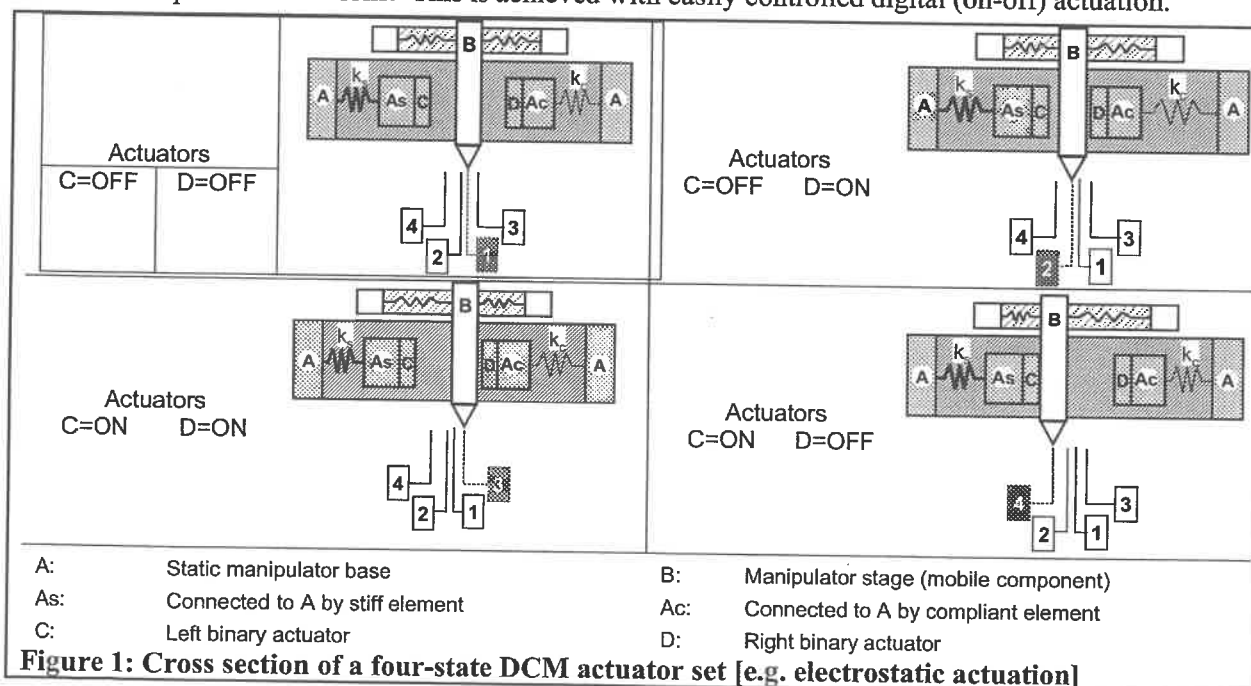
The quality of fixturing, testing/inspection, material handling and other manipulation processes for micro and nano-scale devices depends on the performance of positioning equipment. Many "next generation" positioning technologies are enhanced versions of decades old hexapods, stacked linear stages or flexure stages. Forcing these macro-scale concepts to provide sub-micron performance yields expensive manipulators. These equipment costs are then reflected in micro and nano-scale products, often making up as much as 50% of packaging cost. As a result, many scientists and engineers are now convinced that small-scale products which require manipulation/alignment or assembly can not be commercially viable.

Discrete Nano-Actuation Technology (DNAT) may change this perception. We propose a novel, low-cost manipulator which uses teams of opposed, binary actuators in series with compliant elements to attain many repeatable, discrete positions in a given workspace. The purpose of this paper is to demonstrate the technology via a case study on a macro-scale Cartesian manipulator. The end goal of this research is to miniaturize this concept to a micro-scale manipulator for manipulation of small-scale components (e.g. on-chip Nanomanipulation).

2. DESCRIPTION OF DNAT TECHNOLOGY

2.1. Fundamentals of DNAT manipulation

Figure 1 shows the basic building block of DNAT manipulators, a single, linear DNAT set. We wish to position B (e.g. wave guide or carbon nano-tube) with respect to A using this set. Opposed compliant elements of different stiffness (k_s and k_c) permit sub-components of part A, A_c and A_s , to move relative to A due to compliant connections. This is achieved with easily controlled digital (on-off) actuation.



Actuators C and D, are energized as indicated to produce four positions. It is important to note the range/spacing of states for a set may be set by changing the difference in stiffness between k_s and k_c .

2.2. Practical application of DNAT actuators

The displacements from several sets of DNAT actuators may be superimposed to form a Cartesian (x-y) manipulator. Figure 2 provides a comparison between a single set (4 states) and 3 sets (61 states). The difference in stiffness of each DNAT actuator and the compliance of the manipulator enables one to superimpose different digital actuation combinations to obtain many of the positions within a given work space. Without the difference in stiffness (differential stiffness), attaining many of the states shown in Figure 2 would be impossible. They are combinations of states obtained via differential compliance.

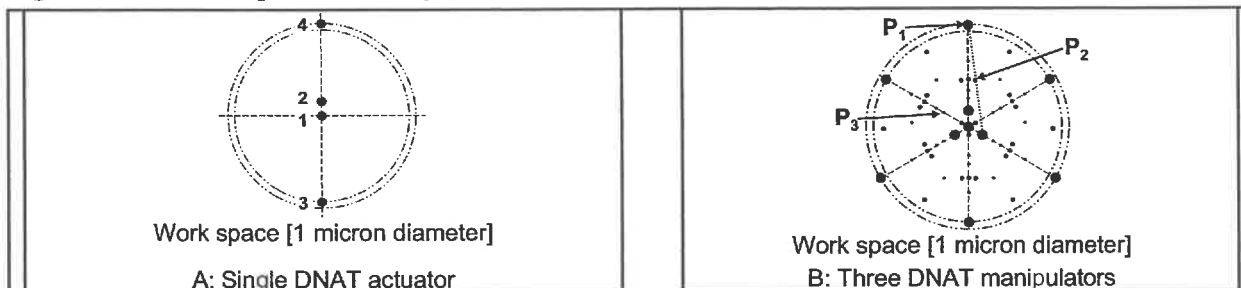


Figure 2: Change in number of positions with number of actuator sets [surface tension actuation]
S = Stiff, C = compliant; Component B is to be positioned with respect to the outer, grounded ring

DNAT devices are used to position at discrete points that achieve an acceptable aligned position. For example, one may move from P_1 to P_2 or to P_3 if this provides an improved state of alignment. By saturating the work space with more points (i.e. adding actuator sets), the performance of DNAT devices approaches that of analog alignment systems. This is easily accomplished as the number of discrete positions scales with: $2^{(2 \times \# \text{ of actuators})}$. Six DNAT sets provide over 4000 positions in a given work space. This yields repeatable positions spaced an average of 15 nm. Smaller or larger spacing may be achieved by changing the difference in stiffness between the opposed compliant elements. The effect of varying the stiffness is shown in Figure 3.

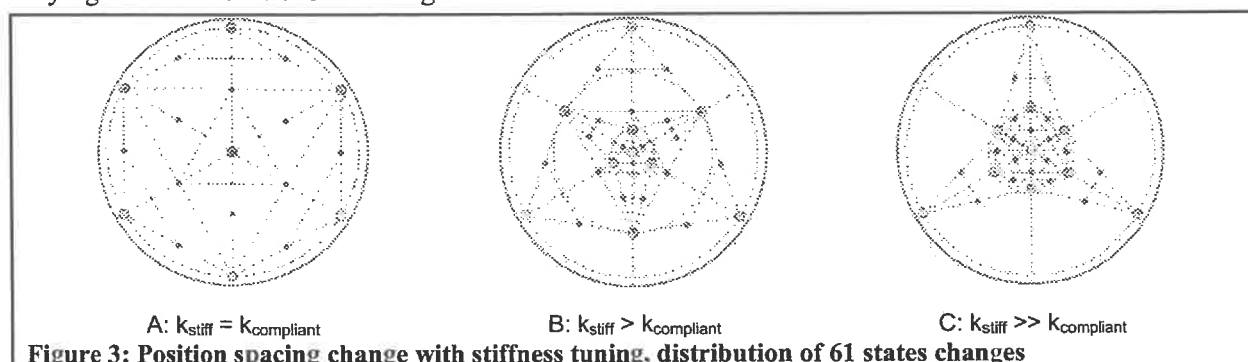


Figure 3: Position spacing change with stiffness tuning, distribution of 61 states changes

2.3. Potential benefits of the technology over traditional precision manipulators

In comparing DNAT technology to traditional systems, one must consider the following:

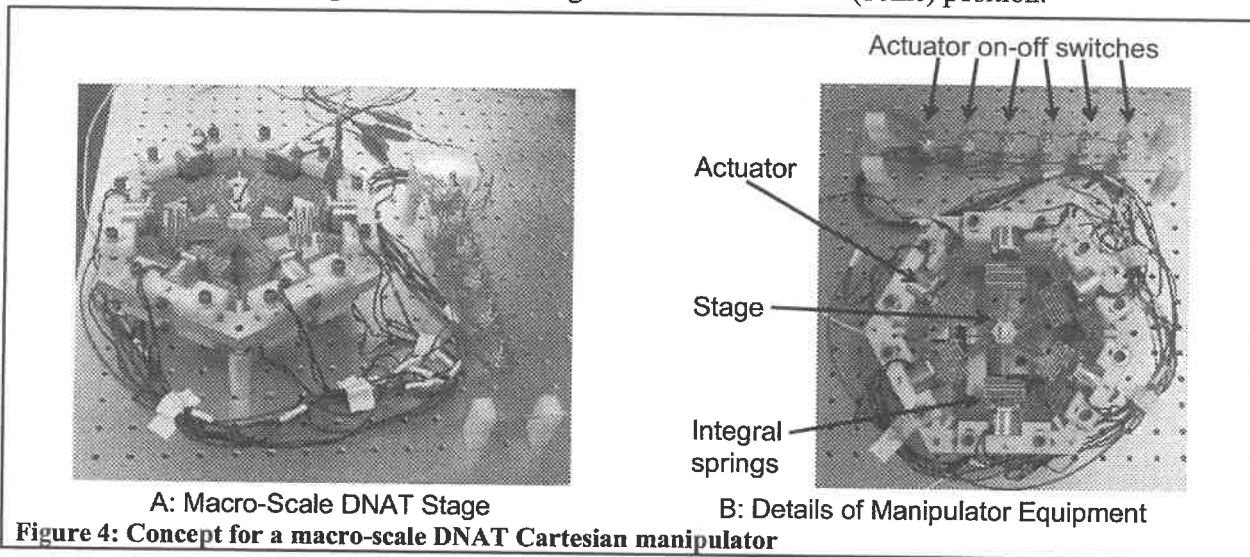
Structure design and manufacture: The manipulator is a monolithic compliant mechanism. Within the mechanism are flexible elements which passively work in teams to provide fine resolution position changes. These structures are easily defined/manufactured via DRIE and other micro-fabrication processes. As a result, integration into existing Microsystems should be possible.

Actuation, sensing and control: Binary actuators provide well-defined (i.e. repeatable) inputs to the manipulator that can be maintained with no power (off position) or low power (on position).

Repeatable binary actuation enables nanometer level repeatability (and accuracy if calibrated/mapped) without extensive sensing/feedback control.

3. EXPERIMENTAL SETUP AND PROCEDURE

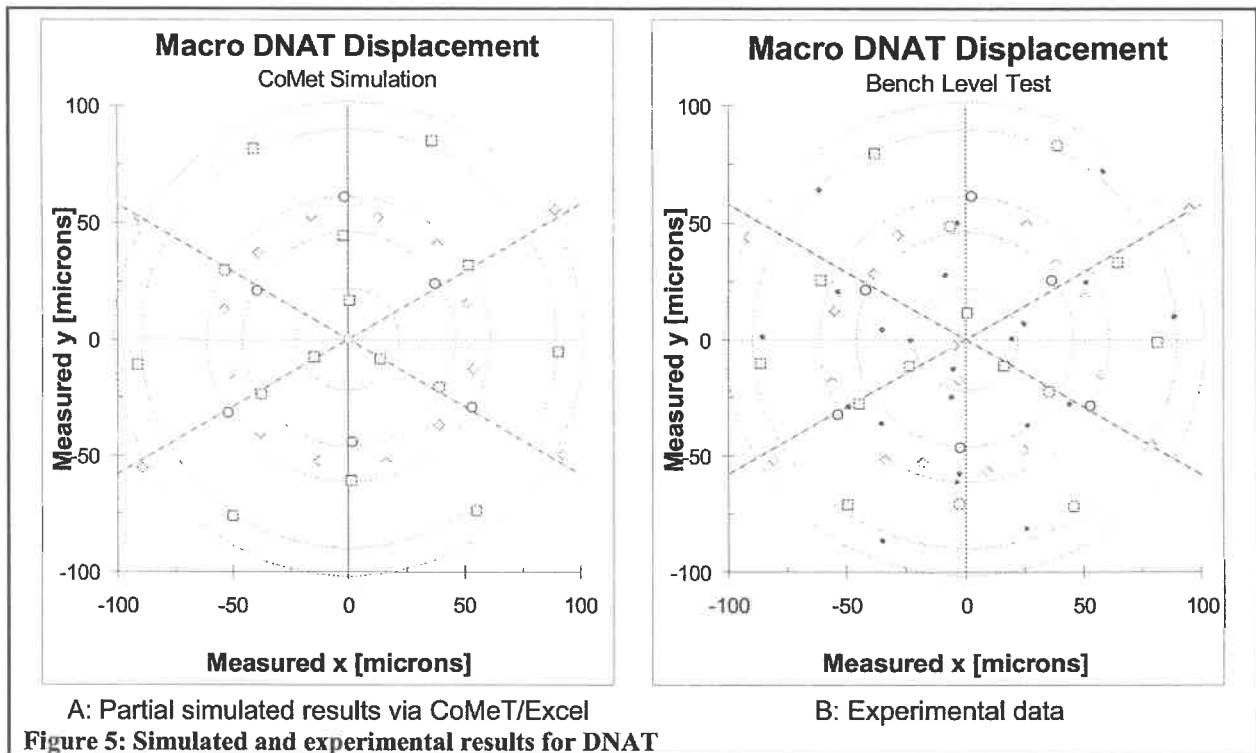
A macro-scale version of a three actuator DNAT manipulator was built to provide a means to characterize the performance of a manipulator concept. Figure 4 shows the macro-scale DNAT manipulator. The heart of the stage is a monolithic compliant mechanism which is cut from Aluminum sheet on an abrasive waterjet cutter. The mechanism is driven by sets of electromagnetic actuators which act in a binary mode (on-off). The electromagnets are thermally isolated from the compliant structure via polymer mounts and stand offs. When energized, they attract and cause a ferrous metal part on the manipulator to contact a hard stop. Thus a well-defined stroke is obtained for each binary actuator. Two capacitance probes are used to read the x and y displacement of the stage from its non-actuated (home) position.



A series of switches (see Figure 4B) were turned on/off to obtain different digital actuation combinations with corresponding unique position states. As indicated earlier, this radial design is capable of producing 61 unique states within a given work space.

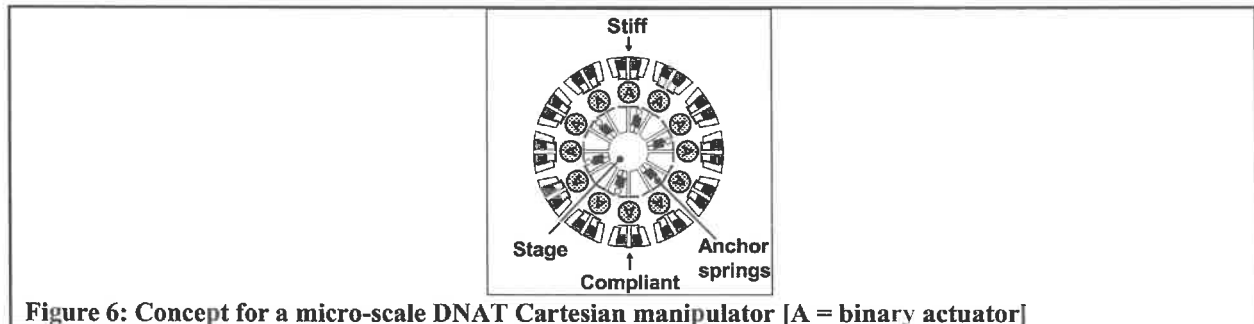
4. RESULTS

Figure 5 shows two plots which compare performance simulation with actual displacement of the DNAT manipulator. The digital actuation combinations and stage displacements were modeled using CoMeT and Microsoft Excel. Figure 5A shows a partial plot of simulated states (not all 61 are shown). Figure 5B shows a full plot of states obtained from an experiment. This comparison shows a reasonable agreement for a first-level prototype. From inspection, it can be seen that most displaced states vary by approximately 20% from their intended position. This is not surprising given the data was taken with no calibration and the fact that the stiffness of the compliant mechanism (local and global) differs from the model due to manufacturing errors (thickness variation and taper from the waterjet cut). Fortunately, the accuracy of this stage can be improved on the macro-scale by calibrating the space (~ 500 microns) between the hard stop and the ferrous target (e.g. shimming). However, on the micro-scale this promises to be more challenging as the final state of the fabricated device may be difficult to move/calibrate to achieve a desired behavior. The repeatability of obtaining a state with the macro-scale DNAT manipulator has been measured at better than 1% of the displaced value.



5. DISCUSSION

The results of the initial test are promising and warrant further study of the technology on the micro-scale. The next step in this research is to focus on building of a base of knowledge on the best way to apply this technology to micro-scale devices. At present, the concept in Figure 5 is being investigated for the first attempt at a micro-scale DNAT manipulator. The manipulator is equipped with on-off electrostatic actuators. Simulations predict this design will attain over 4000 discrete states within a $1\mu\text{m}^2$ work area.



Additional efforts are underway to produce planar, spherical and spatial devices which utilize this digital actuation technology.

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DESIGN OF LOW-COST KINEMATIC COUPLINGS USING FORMED BALLS AND GROOVES IN SHEET METAL PARTS

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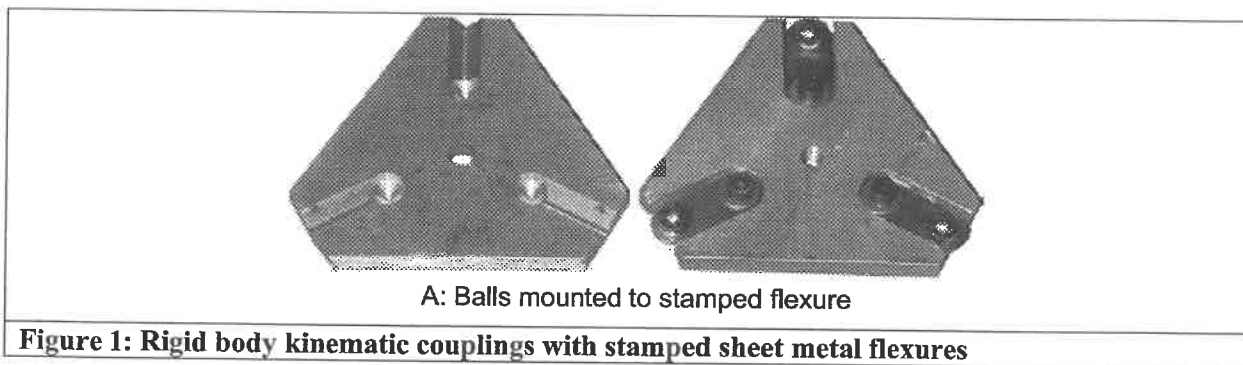
Keywords: Compliant mechanism, Digital actuation, Repeatability, Accuracy

1. INTRODUCTION

The inherent variability in sheet metal parts and assemblies is a problem that has plagued manufacturers for decades. This variation is a result of many causes, the major contributors being:

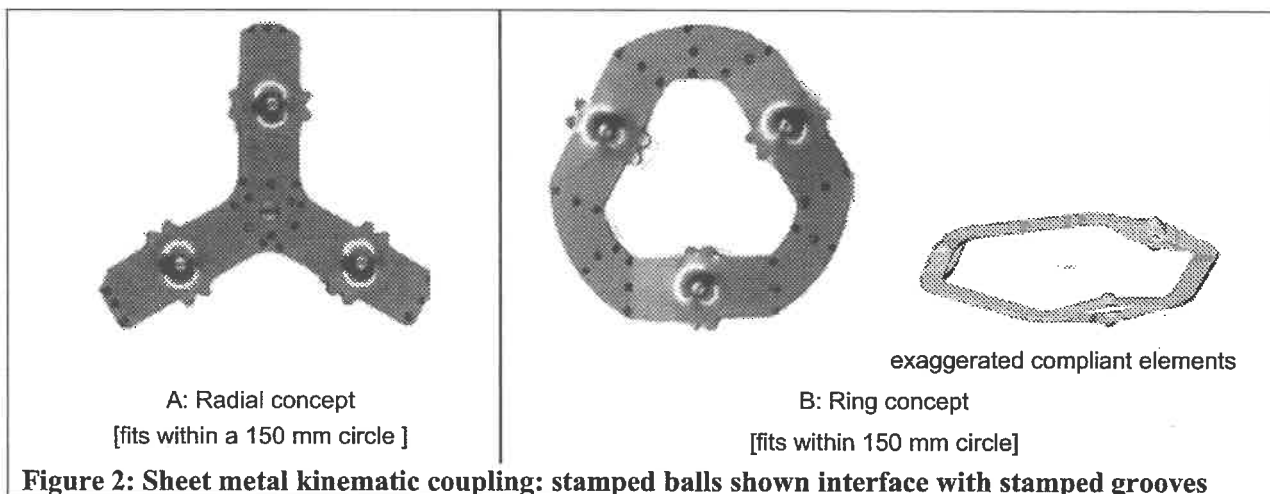
1. Material property variation
2. Variation in the deformation processes (e.g. die-part friction and die-part alignment)
3. Flexibility of assembled parts (error clamp-in)
4. Variation in part-part alignment due to local/global over and under constraint
5. Manufacturing errors in location and size of alignment features

In this work we address the fourth and fifth sources with the intent of minimizing the variation in part-to-part alignment by employing exact constraint techniques in sheet metal assemblies. The goal of this paper is to demonstrate feasibility of this technology through the experimental investigation of sheet metal kinematic couplings. Sheet metal materials have been used by Culpepper et. al [1] to form the elements of precision couplings. These efforts have produced couplings with micron-level repeatability. However, the main goal of this previous work was to use the inherent compliance of the sheet metal to enable adjustment and contact between kinematically constrained components. Two examples of kinematic elements from this work (augmented by sheet metal components) are provided in Figure 1. The design in Figure 1 incorporates a sheet metal cantilever spring to enable sealing contact between the two components.



The goal of the present research is to form kinematic elements within sheet metal components such that they form the major alignment components. Figure 2 shows two prototypes of sheet metal kinematic couplings (ball sides) that were investigated as part of this work. The radial concept is designed for maximum in-plane coupling stiffness (high stiffness in directions perpendicular to v-grooves). The ring

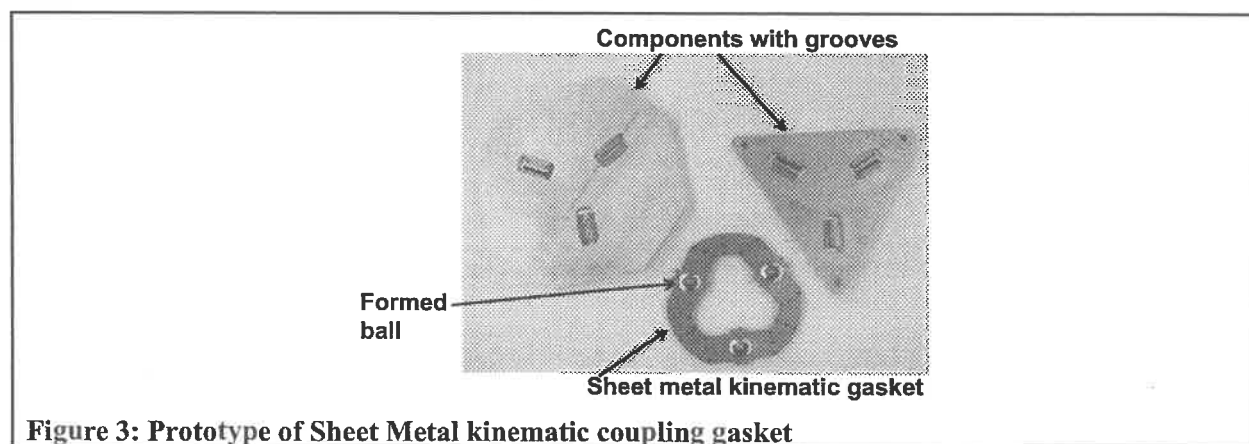
concept is designed with a flexural element (see Figure 2) to enable some relative displacement (stroke) between mated components and perpendicular to the plane of coupling.



This technology can be used to provide product integrated features for part-to-part alignment and/or to act as intermediate parts which align other components. Experimental results from bench-level prototypes show better than 150 nm repeatability for the radial concept [2].

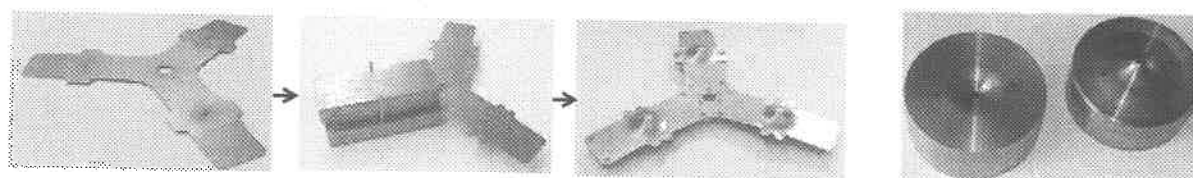
2. DESCRIPTION OF TECHNOLOGY

Figure 3 shows a set of tooling used in the tests for the radial kinematic coupling prototype. The kinematic interfaces shown in Figures 2 and 3 are welded assemblies of two stamped components. Each component has three balls stamped into its surface. The components are then welded such that the balls on each sub-component are aligned and opposed (See Figure 2B, far right). The balls are mated into metal v-grooves which are attached to test components. These components shown in Figure 3 are designed to work with an automated test rig. It should be noted that these components could take on any shape or be formed of other sheet metal components.



The blanks for the kinematic features were cut from sheet metal using an abrasive waterjet cutter. The process of turning a blank into a formed sheet metal kinematic coupling component is provided in Figure 4. The tooling is aligned to the sheet metal blank via a set of spring-loaded pins that fit into v-grooves in

the sheet metal blank. It is important to take care in the choice of blank-tooling alignment mechanism (pins in grooves) and the location of the mechanism. During the forming process, the locating portions of the sheet metal may be pulled/deformed. As a result, registration between the tooling and part may be affected. This is also important if the registration features on the sheet metal are to be used to align to other components, such as to form the welded assembly in Figure 2B. Two methods have been used to avoid or mitigate these problems. First, the spring loaded pins that locate the die with respect blank are preloaded so that they clamp into the v-grooves (see Fig. 4A, middle and right) and thus retain positive registration to the Vs. In addition, one may use standard sheet metal forming methods to clamp registration points/areas so that they may not deform and affect registration. From a performance standpoint, this is preferred, but places constraints on the design of the sheet metal parts (room dedicated to clamping) and requires more expensive tooling (with built in clamping mechanisms). The choice of method (pins or clamping) depends on the application. The forming process used to make the balls in Figure 4 was completed on a manual press in MIT's LMP machine shop.



A: Forming process (from blank to finished piece)

B: Ball forming tooling details

Figure 4: Tooling used to form

3. EXPERIMENTAL SETUP AND PROCEDURE

Figure 5 shows a test kinematic coupling set mounted within an automated test rig. This test rig is equipped with an automated air piston for engaging, disengaging and preloading the coupling set with a 450 N load. Lateral piston forces (those which cause error in coupling plane) are decoupled from the coupling via an isolation flexure place between the coupling and the piston. Six capacitance probes are positioned to read the displacement of the bottom plate (balls) with respect to the top plate (balls). The top plate and capacitance probes are rigidly attached to the test rig frame. A DSpaceTM interface is used to control the air piston (binary on-off coupling preload) and to read data from the capacitance probes.

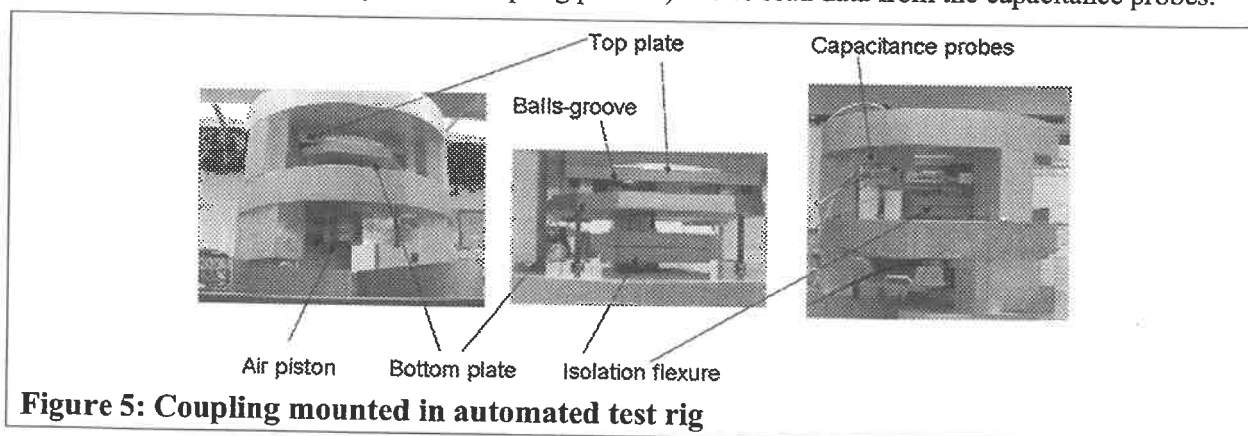


Figure 5: Coupling mounted in automated test rig

The control/data acquisition is programmed to engage the coupling, then allow the coupling components to settle with respect to each other for 30 seconds before data is taken. This procedure is followed to

achieve a stable position equilibrium that occurs after transient settling motions cease. In these tests, the transients could not be sensed after 20 seconds. Once the coupling is settled, 100 data points are taken within one second to average out noise and outliers in the readings. The coupling is then disengaged and the cycle repeats.

4. EXPERIMENTAL RESULTS AND DISCUSSION

The surfaces of the balls and grooves were inspected to ensure the surface finish were comparable. No lubrication was used in either test, however after forming, the sheet metal parts were coated with a hard coating, Titanium Nitride (TiN) to mitigate wear on their soft metal surfaces. Both coupling designs provided better than micron-level repeatability (see Figure 6) once they were worn in. As the friction for each concept should be similar (based upon similar ball-groove geometry, surface finish and coating) it is assumed that the difference in repeatability is primarily due to the difference in coupling stiffness. In fact, FEA predicts that the difference between the in-plane stiffness of the two designs varies by a factor of 100%. This is close to the variation in repeatability of 80% with the balance of the difference likely due to surface finish/friction effects. This first-level demonstration of sheet metal kinematic coupling technology feasibility warrants further investigation.

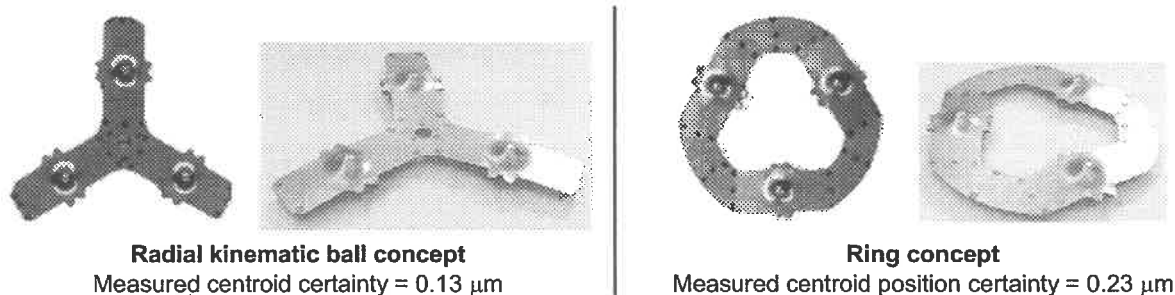


Figure 6: Sheet metal kinematic coupling: Stamped balls shown interface with stamped grooves

Future research efforts are focused on developing thin-film contact mechanics and compliance matrices that can be used to form deterministic models of coupling stiffness. Statistical models are being developed which predict coupling accuracy as a function of forming variation and tolerances on the size/location of the balls and grooves. Research in the application of this technology to Ford sheet metal measurement (e.g. CMM interfaces) and assembly (between sub-components of doors) are underway.

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Precise manipulation control on three versatile micro robots for flexible micro handling

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1. Introduction

In MEMS technology and precision engineering, there have been developed many reports about micromechanisms and microrobotics. Some are based on advanced technology including microbatteries, micromotors, and miniscule computational facility[1], the others are made by sophisticated precision machining techniques[2][3]. It is still very important to find industrial applications where such microrobots can provide effective benefits. For these years, our group has developed insect size microrobots equipped with microtools and instruments[4]. In-situ micro processing under microscopes is one of interesting application for microrobots[5][6]. For these years, we have developed versatile micro robot for the microscopic manipulation[7][8]. In this paper, we propose flexible micromanipulation organized by three versatile microrobots under microscopes. In order to realize the micromanipulation by the versatile microrobots, we developed the spherical micro manipulator as a microscopic manipulator. In experiments, we demonstrate precise, flexible handling of a minuscule pipe under the good collaboration of these small robots.

2. System Configuration

Fig.1 shows the flexible micromanipulation by versatile microrobots under microscopy which is ongoing development in our laboratory. Operators can control versatile microrobots using a PC in real-time monitoring of microscopic images. All microrobots are set on a steel table. In this basic setup, the operators can execute flexible microscopic tasks with easy operation. All positioning facilities are given by microrobots' movement so all mechanical functions are simply divided. This unique arrangement allows the system good flexibility and high mechanical stability although sophisticated control is required. This may be a good application for microrobots practically. We can easily attach microrobots to microprocessing instruments. This

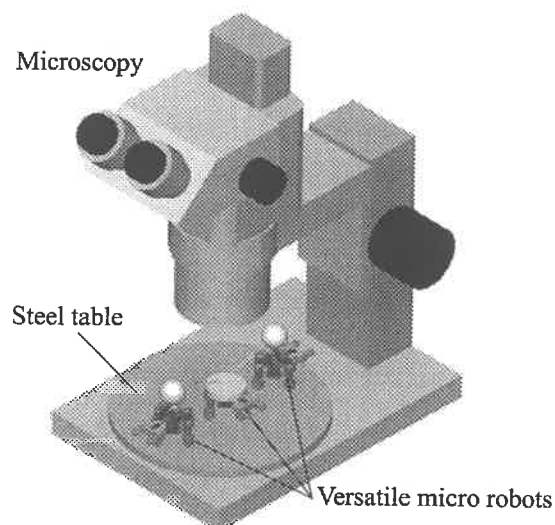


Fig.1 Versatile micro robots for flexible micromanipulation under microscopy

flexibility meets the requirements of users, and the use of small robots avoid reparation of microscopes and easily implements processing with low cost.

3. Versatile micro robot

Fig.2 shows motion patterns required at microscopic operation. To realize these motion patterns at the same time, we should design the structure which can move in XY directions and in rotation independently. Fig.3 shows the structure of the versatile microrobot which is proposed to satisfy the requirement above. Two U-shaped electromagnets which are arranged to cross each other are connected by four piezo elements so that the microrobot can move in any direction with the manner of inchworm. Also we design the special joint at one of 4 legs to get 4 legs smooth contact on the surface simultaneously. In experiments, we confirmed that the microrobot can move in XY directions and in rotation independently with nanometer resolution at different

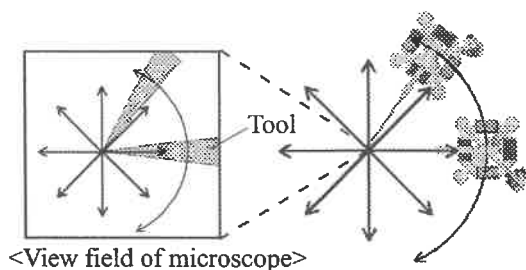


Fig.2 Motion pattern required at microscopic operation

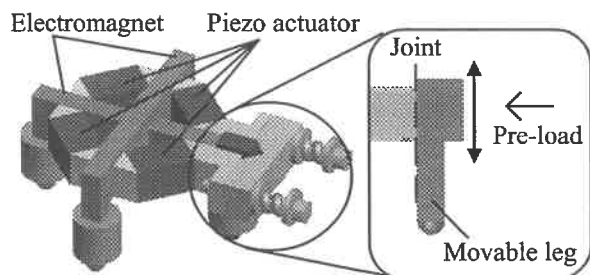


Fig.3 Structure of versatile microrobot

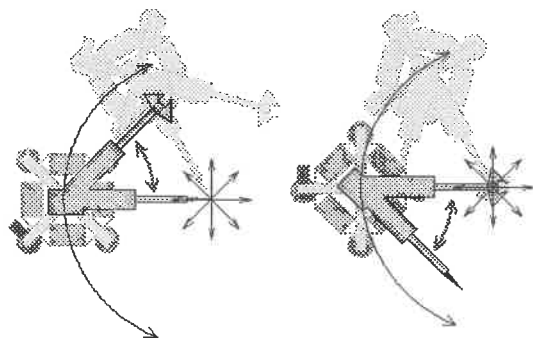


Fig.4 Flexible positioning of several tools by one robot

positioning speed[7]. We also succeeded to compensate for the motion very well[8]. As an interesting function, the robot can position several tools without extra manipulator as shown in Fig.4. We believe this unique function is effective for the advanced microscopic tasks at which we need several tools.

4. Spherical micro manipulator

4.1 Structure and drive sequence

We design the spherical micro manipulator as a microscopic manipulator as shown in Fig.5. The manipulator is composed of three piezo elements with a small accurate sphere and a large accurate sphere on the top of them. Three piezo elements are arranged at the apexes of a regular triangle and support the rotation sphere as shown in Fig.6 to allow the stable contact at any angle. When we apply saw-tooth waves to three piezo elements, the sphere rotate around on AB axis

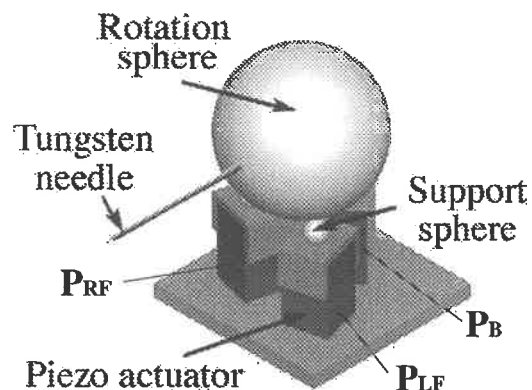


Fig.5 Structure of spherical micro manipulator

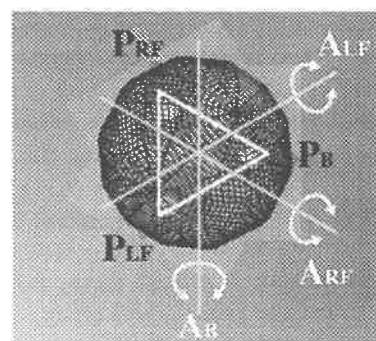


Fig.6 Arrangement of three piezo actuators (top view)



Fig.7 Spherical micro manipulator provided for microscopic manipulator

precisely based on an stickslip model. When the similar sequence is operated, the sphere rotates around on AFR and AFL axes. The electropolishing tungsten needle is attached to the large sphere as shown in Fig.7, and we can control the tungsten needle in Z direction as well as the angle precisely.

4.2 Experimental results

As shown in Fig.7, we fabricated the spherical micro manipulator to check its primary performance. We use the brass sphere for support spheres, the stainless sphere for rotation sphere and stacked type PZT elements of 5mm x 5mm x 10mm. In order to check the angular

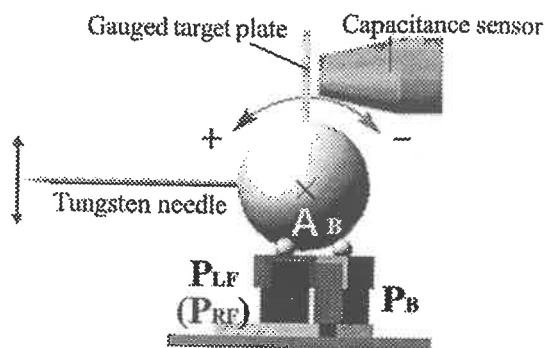


Fig.8 Experimental setup of rotation measurement around on AB axis

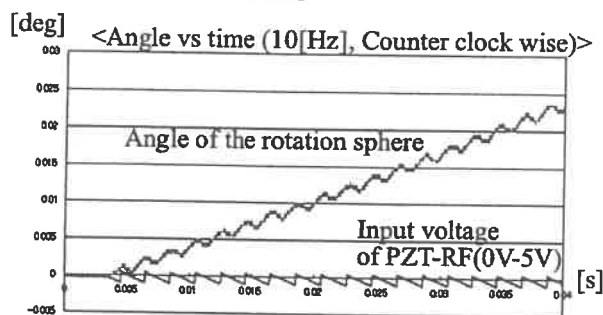


Fig.9 Relation between angular displacement and input voltage

displacement of the rotation sphere by a capacitance sensor, we attach a metal plate to the rotation sphere as shown in Fig.8. Fig.9 shows relation between angular displacement and the input signal of the saw-tooth wave. It is found that the sphere rotate 8.5×10^{-4} degrees per one step when the amplitudes of saw-tooth waves are 5 voltage. Hence this means that the end of the tungsten needle which length is 50mm move with the resolution of 0.7 micron per one step in Z direction. We have also confirmed that the angular speed is proportional to frequency very well up to 1kHz. When the drive frequency of 1kHz is given, then the tungsten needle moves 700 micron per one second in Z direction. In these experiments, we confirmed that the spherical manipulator has sufficient angular resolution and angular speed as a microscopic manipulator.

5. Micromanipulation Organized by Three Versatile Microrobots

5.1 Collaboration of Three Robots

Fig.10 shows the layout of micromanipulation by cooperation of three versatile microrobots. We attach the spherical micro manipulator to two versatile microrobots to handle micro samples in cooperation with each other. We also arrange another versatile microrobot with a sample stage under a microscope. Tungsten needles are positioned in X, Y, and rotational direction by the versatile microrobot and positioned in Z direction by

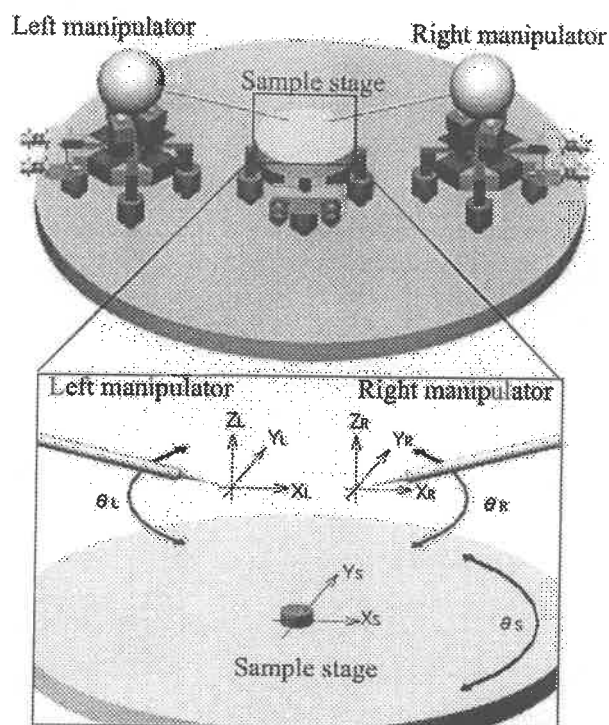


Fig.10 Layout of micromanipulation organized by three versatile microrobots

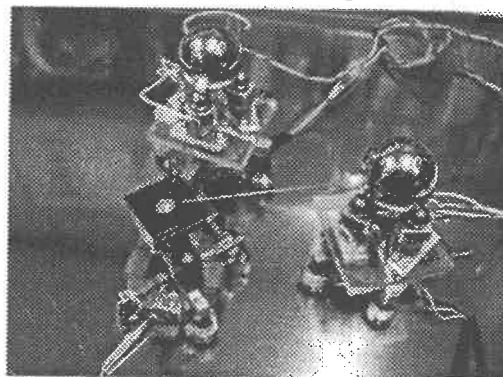
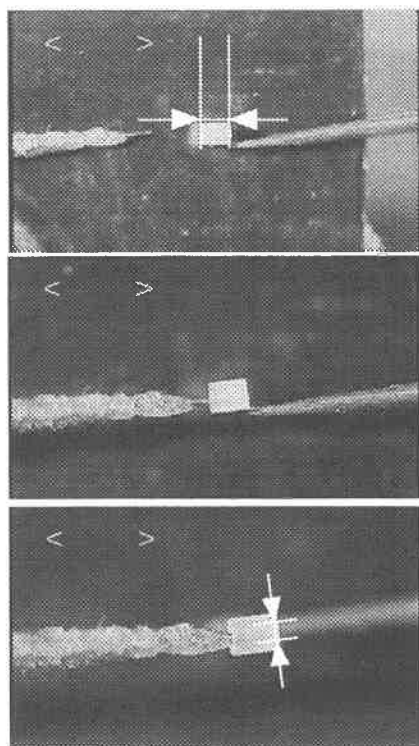


Fig.11 Flexible micromanipulation by collaboration of three versatile microrobots

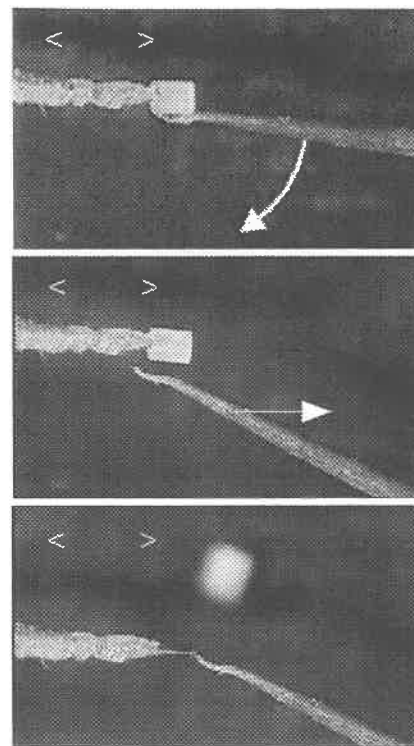
the spherical micro manipulator. The sample stage is positioned in X, Y and rotational direction by the versatile microrobot. Both tungsten needles have 4 DOF and the sample stage have 3 DOF, so the manipulation system has 11 DOF. Because the positioning range of our microrobots is only limited by the steel table, the robot positions its manipulator at any point in the steel table. This arrangement of three versatile microrobots provides both flexible and accurate micromanipulation with nanometer resolution.

5.2 Experimental Operation

Flexible micromanipulation organized by three versatile microrobots are arranged for primary experiments as shown in Fig.11. We also developed the special control software so that each motion could be



<Inserting a needle into a pipe>



<Removing a pipe from a needle>

Fig.12 Flexible handling of a miniscule pipe by two tungsten needle

controlled by a simple mouse click and a joystick on the PC for primary experiments. The locomotion frequencies of microrobots are adjusted from 0 to 500Hz to select optimal speed according to magnification of the microscopic image. We succeeded in inserting a tungsten needle into the miniscule pipe and removing it by cooperation of two manipulator robots as shown in Fig.12. We believe that we can apply this performance to operations of biological samples and microdevice repair.

6. Conclusions and Future Works

Flexible micromanipulation organized by versatile microrobots under microscopes was proposed and developed to show the feasibility of microrobots. The demonstrations of primary micromanipulation assisted by the microrobots were described and some unique tasks were achieved. We are now improving the PC controlled system based on visual feedback to make it more precise and reliable. To realize more advanced and complicated work, we are improving the spherical micro manipulator and we are developing the software to control the several tools by one robot. We plan to apply our system to biomedical applications and in-situ micromanipulation in scanning electron microscopy.

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Micro Turning System 3: A Super Small CNC Precision Lathe for Microfactories

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Introduction

Micro Turning System 3 (MTS3) is a micro precision lathe that has a machine base size of 300 x 200 mm (A4 paper size) and PC (personal computer) based CNC (computerized numerical control) with linear/circular interpolation. The machine was developed to be a substitute of conventional precision lathes that were too big compared to the size of the parts to be machined. MTS3 is capable of machining stainless steel (304) with a surface roughness of 60 nm (Ra) and a circularity of 50 nm (P-V). Figure 1 shows the picture of MTS3.

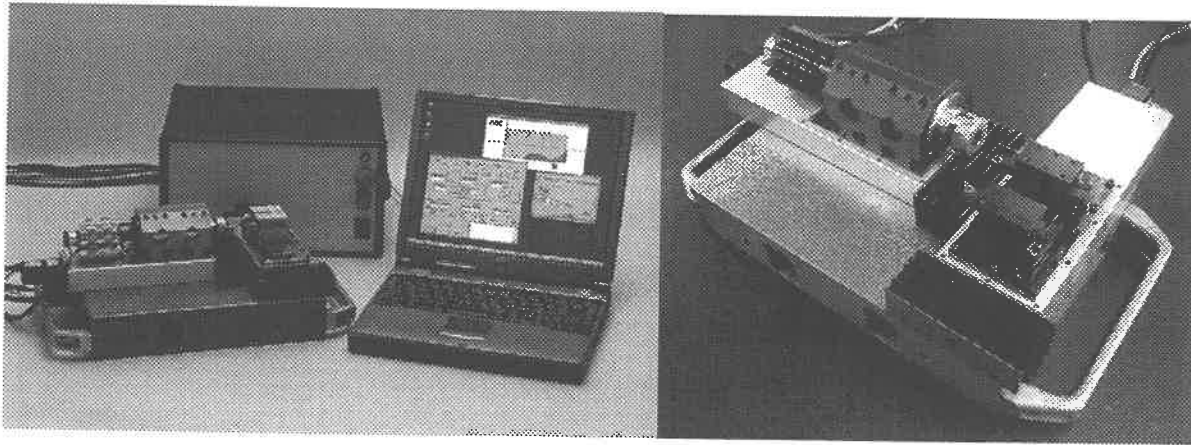


Figure 1: The whole system and close-up view of MTS3.

Background

There has been a nebulously defined concept that tooling machines has to be large enough to perform a precision machining, which has prevented the development of small tooling machines. However, the recent advances in high-precision processing technology made micro tooling machines feasible. There are several reasons of having micro tooling machines.

- 1) Micro machines save a large amount of energy. The use of over-sized machines for small parts results an over consumption of energy that have been ignored; however, the market today requires further efficient ways of manufacturing. Lower energy consumption will be obtained by utilizing smaller machines with the parts' size becomes smaller. Suppose a conventional tooling machine consumes 10 kW of electricity, a micro machine powered 100 W is more efficient by 100 times.

- 2) Micro machines save the cost for peripheral equipment such as temperature control or vibration isolation. In order to perform a precision machining, the temperature of a machine has to be maintained constant; that is, an entire room temperature is to be controlled for the machine. For a micro machine, on the other hand, such temperature-controlled room is not necessary; rather, a small chamber is enough to control its temperature. Vibration isolation is also an important factor to make a precision machining. A large machine needs a vibration-free machine bed or even special building to isolate internal/external vibrations. A micro machine may not need to have such system because it difficult to be affected by disturbance: micro machines have higher natural frequency than the frequencies of vibrations caused by machining or surroundings.
- 3) Micro machines will let manufacturers have more flexible factory layout, and they can easily be rearranged their layout in response to the changing market. Because a micro machine does not need a high-powered electric source, special room or building, they can be installed in any place or can be moved anywhere with low cost.

Main Features of MTS3

MTS3 was developed as a larger version of MTS2, a palm-top lathe that was introduced in the 2002 ASPE Annual Meeting. The size of the machine was enlarged in order to make MTS possible to turn harder materials and to have multi-tool installation. The specifications of MTS3 are listed on the Table 1.

Machine Base Size		300 x 200 mm
Maximum Cutting Length		50 mm
Maximum Chucking Diameter		10 mm
Main Spindle	Reducer Type	Traction Drive
	Maximum Speeds	3,000 rpm
	Radial Rigidity	20 N/ μ m
	Axial Rigidity	35 N/ μ m
	Run-Out	2.0 μ m
X/Z-Axes	Slide Type	Cross Roller
	Feed Type	Slide Screw (0.2 mm pitch)
	Table Stroke	55 mm
	Rapid Traverse	0.38 m/min
	Smallest Feed	100 nm
	Radial Rigidity	180 N/ μ m
	Thrust Rigidity	30 N/ μ m
	Positioning Accuracy	420 nm
Motor for Main Spindle	Straightness / Travel	350 nm / 40 mm
	Type	AC Servo
	Rated Output	10 W
	Rated Torque	16.4 mN-m
Motors for Axes	Max. Torque	40.2 mN-m
	Type	AC Servo
	Encoder Resolution	2048 pulse/rev
	Rated Output	10 W
	Rated Torque	31.8 mN-m
	Max. Torque	95.5 mN-m

Table 1: Specifications of MTS3.

X and Z-Axis

Both X and Z-axes are driven by 0.2-mm-pitch threads and have cross-roller ways for their slide. Each motor for the axes is AC servo that has a power consumption of 10 W and a 2048-pulse-per-revolution encoder. The encoder signal is feed backed in semi-closed loop to control each axis with a smallest feed of 100 nm. The repetitive positioning accuracy of axes was measured as 420 nm, and the straightness was measured as 350 nm for 40-mm travel. Figure 3 shows the error of repetitive positioning for 10-mm feed.

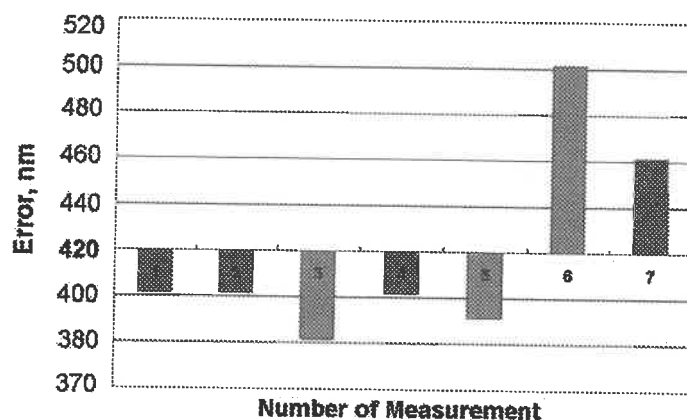


Figure 2: Repetitive positioning accuracy of X/Z axis for 10-mm feed.

Main Spindle

The main spindle consists of a 10-W AC servomotor, a traction-drive planetary roller reducer, a set of angular ball bearings, and a collet-chuck that has the maximum chucking diameter of 10 mm. The ball bearings are chosen because they do not need any extra power. The planetary roller reducer, which has a speed reduction ratio of 3:1, is installed to increase the motor torque. The traction drive was chosen over planetary gears because the traction drive made fewer vibrations and was fabricated at lower cost. The size of the spindle is 50 mm in diameter and the maximum speed is 3,000 rpm after speed reduction.

Control System

The motor control system is consisted with a set of motor driver and controller, a NC board, sensor amplifiers, a 12-volt-DC power source, and a PC. Tool-path is generated on the PC with Windows 98/2000, and the data is transmitted to the NC board via USB (universal serial bus) interface. Users can create the NC data using drawing feature of the program, directly inputting coordinate data, or inputting G-code system.

Test Cutting and Results

Stainless steel (304) and some other materials were chosen for test cutting. Turning a 5-mm-diameter SUS304 bar with a cutting speed of 31.4 m/min, we obtained a surface roughness of 60 nm (Ra) and circularity of 50 nm (P-V). Figure 3 shows pictures of various materials turned with MTS3. Figure 4

shows the picture of SUS304 bar that its circumference was turned, and Figure 5 and 6 shows the print out of measured surface roughness and circularity of the SUS304 bar on Figure 4 respectively.

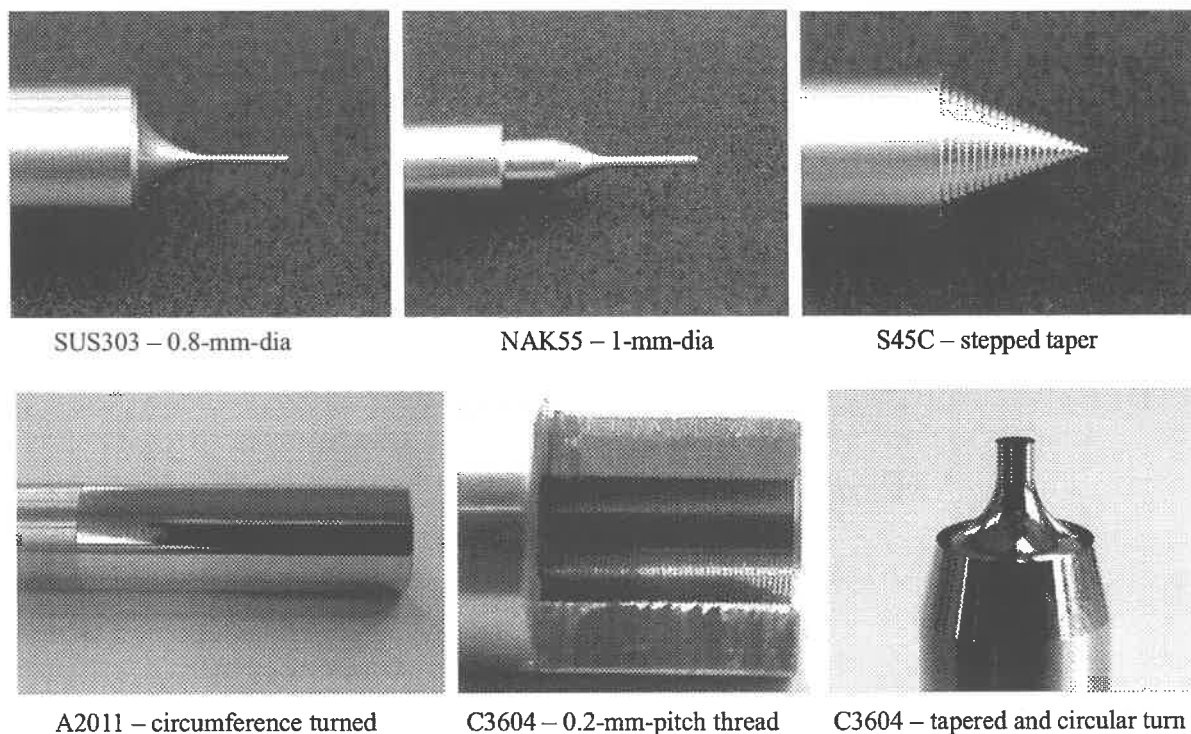


Figure 3: Pictures of various materials turned.

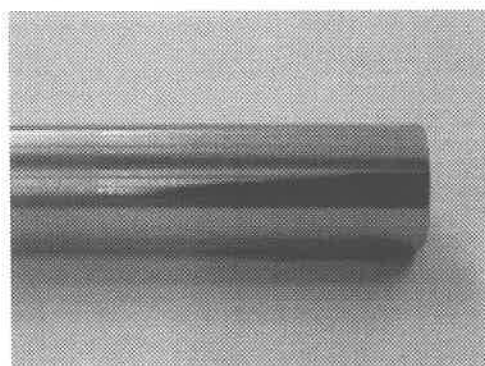


Figure 4: SUS304 – circumference turned.

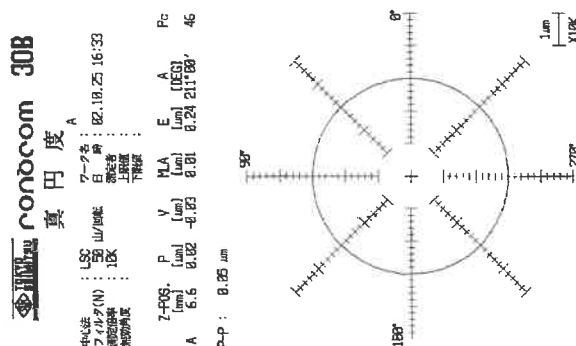


Figure 5: Circularity of SUS304 bar on Figure 4.

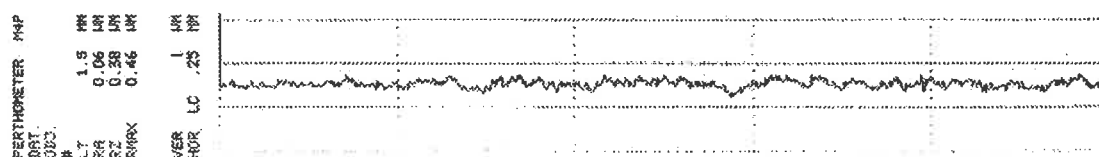


Figure 6: Surface roughness of SUS304 bar on Figure 4.

AN INCHWORM TYPE IN-PIPE MOBILE MICROROBOT DRIVEN BY THREE GAS-LIQUID PHASE-CHANGE ACTUATORS

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1. Introduction

New in-pipe microrobot which inspect or repair thin pipes in the human body or pipelines have been searched. Pneumatic pressure or water pressure have been utilized for the power of in-pipe mobile microrobots [1], [2]. However, we noticed the power of the large change in volume that occurs during a phase-change between liquid and gas [3], [4]. When a liquid is vaporized, the volume of phase-changed gas is more than 100 times that the liquid. The gas-liquid phase-change actuator is a kind of thermal engines that need thermal energy. These actuators are suitable for microactuators used in microrobots in thin for thin pipes in the human body or pipelines, because conventional electrical power or laser beams conducted by optical-fibers can be used as the sources of vaporizing energy. We fabricated an inchworm type in-pipe mobile microrobot driven by three gas-liquid phase-change actuators. The actuators are made of welded stainless steel bellows 7.8 mm thick in outer diameter. The operating fluid is a perfluorocarbon, because its vaporizing temperature is higher than the human body and its latent heat for vaporization is only 1/22 that of water. Consequently, the energy needed for vaporization is very low. The fabricated microrobot is confirmed to move stably with the speed of 2.4 mm/s.

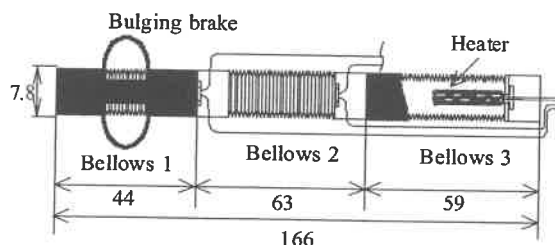


Fig. 1 Structure of the microrobot

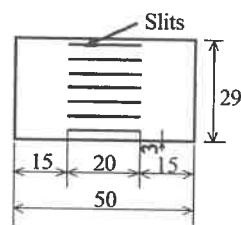


Fig. 2 The bulging brake

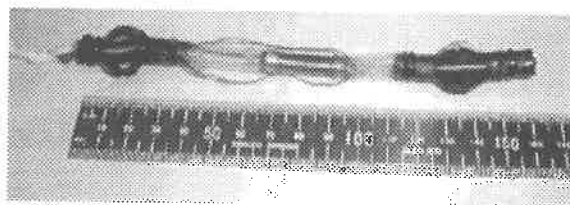


Fig. 3 A photograph of the fabricated microrobot

2. Structure of the fabricated microrobot

Structure of the fabricated in-pipe mobile microrobot is shown in Fig. 1. The microrobot consists of 3 gas-liquid phase-change actuators. The actuator is made

of welded stainless steel bellows whose dimension is 7.8 mm in outer diameter, 3.0 mm in inner diameter. The operating fluid and a heater are enclosed in the bellows. The heater to vaporize the operating fluid is made of a coiled constantan wire which is 0.2 mm thick and 320 mm long. Its electrical resistance is 5.2 W. The operating fluid is non-chlorine perfluorocarbon ($C_5F_{11}NO$). Its vaporizing temperature at atmospheric pressure is 325 K (122 degree in Fahrenheit) and the latent heat for vaporization is 104.56 kJ/kg.

The second (center) actuator 31 mm long is called as a driving element, because the actuator drives the inchworm type microrobot by its stretching and shrinking motion. The first (front) and the third (rear) actuators 44 mm long are called as holding elements, because the actuators are provided seven frictional bulging brakes. The frictional bulging brake which is shown in Fig. 2 is made of vinyl chloride or nitrile butyl rubber (NBR). The holding element works as a brake of the inchworm type microrobot. Friction force of vinyl chloride is smaller than that of NBR. When the actuator of the holding element shrinks, the diameter of seven frictional bulging brakes is 18 mm. The spread bulging brakes surely hold the pipe. According to the stretching of the actuator, the diameter of seven frictional bulging brakes decreases. When the actuator of the holding element stretches 3 mm, the diameter of seven frictional bulging brakes is 10 mm. Then the braking is canceled. The photograph of the microrobot at the free condition is shown in Fig. 3. The microrobot control system is shown in Fig. 4. The system consists of a computer, a power supply and an electric power controller. The cycle of the electric power is shown in Fig. 5.

3. Moving principle

Moving principle is shown in Fig. 6.
(a) Step 0: All actuators are shrinking.

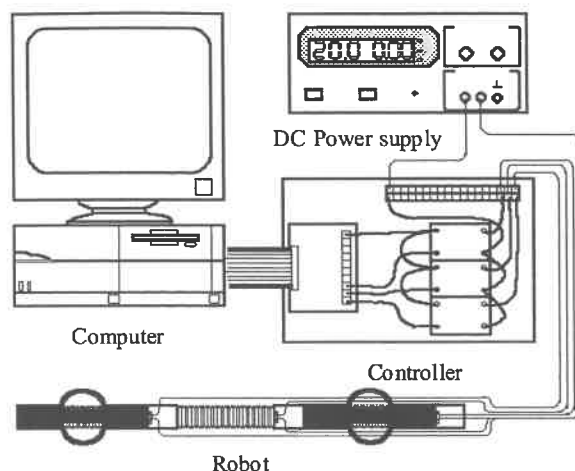


Fig. 4 The control and driving system

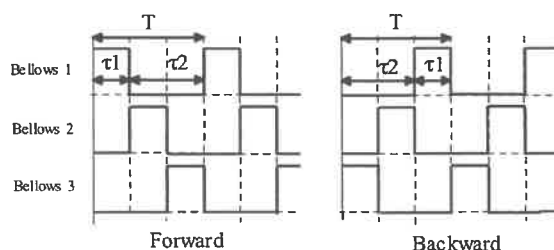


Fig. 5 Cycle time for the microrobot

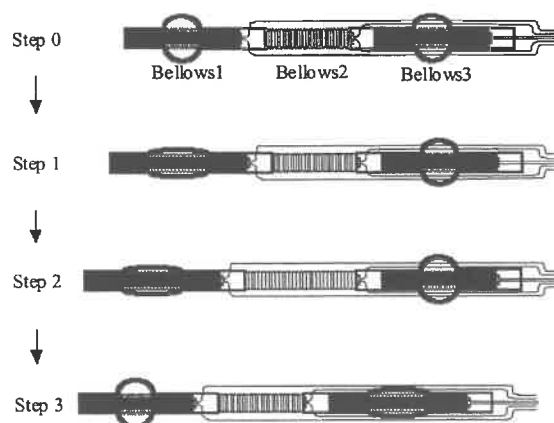


Fig. 6 The principle of microrobot

Consequently, the diameter of frictional bulging brakes of the first and third actuator is large and the braking is done.

(b) Step 1: Electrical power is supplied to the first actuator. Then the actuator

stretches and the frictional bulging brake is released.

(c) Step 2: Electrical power is supplied to the second actuator. The actuator stretches and pushes out the microrobot. At this period, electrical power to the first actuator is cutting off. However, the frictional bulging brake is releasing, because the gas is liquefying and the actuator is stretching. Consequently, the front end of the microrobot is pushed out.

(d) Step 3: Electrical power is supplied to the third actuator. The actuator stretches and the frictional bulging brake is released. At this period, the first actuator already shrinks. The diameter of frictional bulging brakes of the first is large and the braking is done. Electrical power to the second actuator is cut off and the actuator shrinks. Consequently, the rear end of the microrobot is pushed out to the forward direction.

(a) Step 0: All actuators are shrinking. Consequently, one cycle of stretching and shrinking motion of the microrobot is finished. The microrobot is moved to the forward direction in one cycle. If the cycle is reversed, the microrobot can move to the backward direction.

4. Moving experiment

The microrobot is inserted in a 16 mm inner diameter pipe filled with water. The moving characteristics of the microrobot equipped frictional bulging brakes made of vinyl chloride is shown in Fig. 7. When three actuators are stretched in turns from the first to the third, the microrobot moves to the forward direction. When the order of stretching in three actuators is reversed, the microrobot moves to the backward direction. However, the difference of speed between the forward direction and the backward direction is small. Maximum speed of 2.4 mm/s was obtained at the cycle time of 1000 ms.

The moving characteristics of the microrobot equipped frictional bulging brakes made of NBR is shown in Fig. 8. The

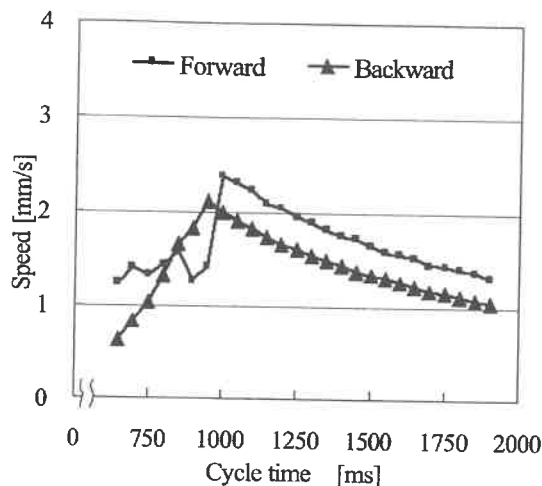


Fig. 7 Moving speed by the vinyl chloride brake

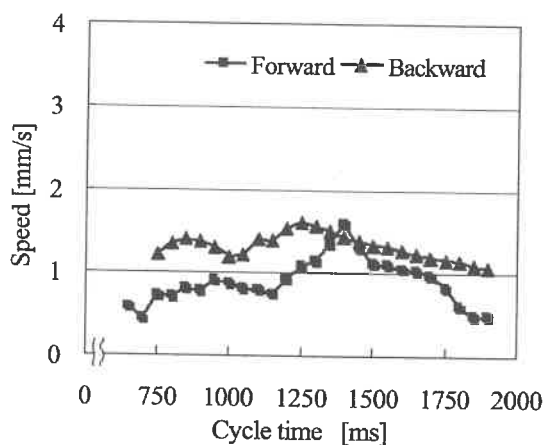


Fig. 8 Moving speed by the NBR brake

difference of speed between the forward direction and the backward direction is also small in this case. Maximum speed of 1.6 mm/s was obtained at the cycle time of 1300 ms. It is confirmed that the speed of microrobot equipped brake of small friction coefficient is larger than the speed of microrobot equipped brake of large friction coefficient.

The speed of the microrobot may be improved by decreasing of the operating fluid and refinement of the frictional brakes. The

microrobot may be used for the heart catheter, if we apply thin bellows actuators less than 1 mm.

5. Conclusions

(1) We fabricated an inchworm type in-pipe mobile microrobot driven by three gas-liquid phase-change actuators.

(2) We confirmed that the microrobot moves to the forward direction when three actuators are stretched in turns from the first to the third and the microrobot moves to the backward direction when the order of stretching in three actuators is reversed.

(3) The microrobot was inserted in a 16 mm inner diameter pipe filled with water. Maximum speed of 2.4 mm/s was obtained by the microrobot equipped frictional bulging brakes made of vinyl chloride.

References

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- [4] S. Kato and T. Shintani, "Fabrication and Thermal Analysis of a Bellows type Gas-liquid Phase-change Micro-actuator", *Proceedings of ASPE 1997 Annual Meeting*, (1997), pp. 315-318.

Structured Surfaces Damping Enhancements at Mechanical Interfaces

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The development of high-power, high-speed machine tools has lead to a practical need to predict, enhance, and control regenerative chatter. While techniques such as tool tuning have been used to significantly enhance machining stability in high-speed machining, increasing the damping is a more effective means of enhancing the overall stability of the machining system. The damping in machine structures, however, is extremely difficult to predict or control because it depends strongly on the microscopic contact conditions at the mechanical interfaces in the system. Our objective is to design structured surfaces aimed at intentionally creating contact that enhances the damping and the dynamic repeatability of the mechanical connections in a machine tool without compromising the kinematic repeatability and accuracy. This will result in increased stability, productivity and machine utilization.

Previous work¹ on the characterization of the machine tool spindle to toolholder connection showed that decreased drawbar load and worse fit actually yielded a system that was dynamically stiffer and more resistant to regenerative chatter. It was also found that interface damping is difficult to predict or repeat from tool to tool, and that what would traditionally be considered a "good" fit between the tool holder and spindle might not be "good" for dynamic stiffness (the product of stiffness and damping ratio) and dynamic repeatability. The explanation for this counter-intuitive result was that dynamic stiffness actually increased with decreased drawbar load; the decrease in stiffness was more than offset by increased damping over a certain drawbar

load range. It is believed that this allowed relative motion (on a microscopic scale) between the two contact surfaces (Rivin²) resulting in an increased damping ratio without unduly compromising stiffness and tool position accuracy. From this research came the following question: Is it possible to use engineered surfaces to decrease the sensitivity of dynamic performance to interface tolerances and increase interface damping? Some preliminary evidence indicates that the answer is "yes", but physical mechanisms remain unclear. If it succeeds, this work would apply to a broad range of structures and precision devices for which damping is important, including aircraft, automobiles, heavy machinery and precision high-speed machine tools.

In this research, a toolholder was tested in an instrumented spindle test stand to find a base line frequency response function (FRF – the ratio of output to input as a function of frequency). The tapered surface was intentionally altered on a millimeter scale and retested. Figure 1 shows the surface after alteration.

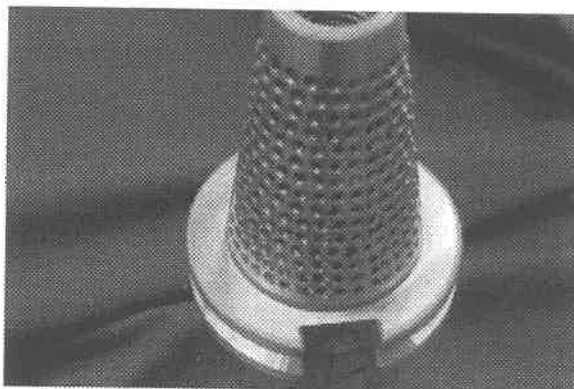


Figure 1: CAT-40 tool holder with altered taper surface

Comparing the two FRF's showed a significant decrease in natural frequency and magnitude at resonance. Additionally, the form of the FRF of the altered taper suggests the presence of nonlinearity.

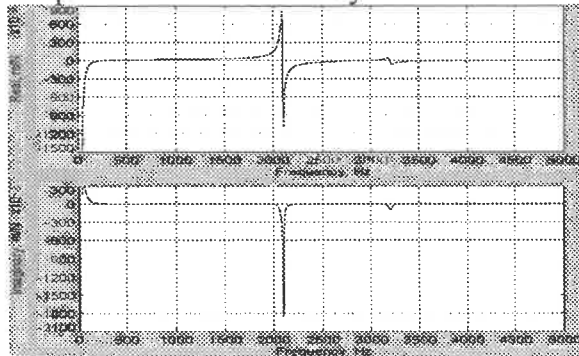


Figure 2 (a): Real part of baseline FRF

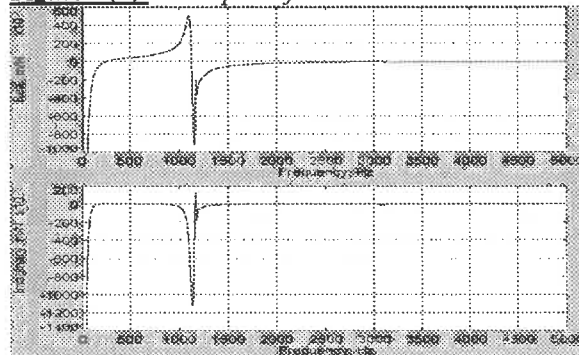


Figure 2 (b): Real part of FRF with altered contact surface

The hypothesis that this nonlinearity may be hysteretic behavior expected due to the micro-scale surface slip (dry friction) is still under investigation. If this explanation is true, we would expect the nonlinearities to be evident in the dynamic testing since dry friction is a nonlinear process. To check for the presence of nonlinearity, the structured toolholder developed by Smith and Jacobs was retested using hammer impacts with a controlled force level. Figure 3 shows the real and imaginary parts of the FRF of the toolholder spindle combination for a very low maximum impact force level of

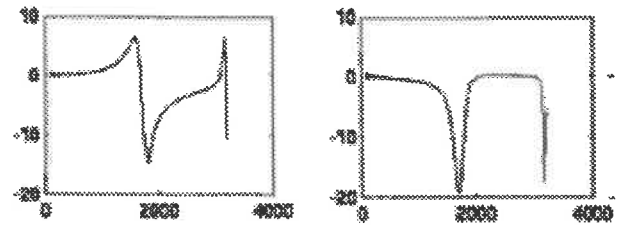


Figure 3: Real and Imaginary parts of FRF of the altered toolholder for impact force of $95\text{N} \pm 5\text{N}$

$95\text{N} \pm 5\text{N}$. This shows a classic linear vibration response: The real part is nearly symmetric and changes sign at the natural frequency, and the imaginary part has a smooth symmetric peak at the natural frequency. In impact testing, nonlinearities cause the frequency response function to change with changes in force level; in a linear system, the FRF is independent of the force level. Figures 4 (a) and (b) show the real and imaginary parts of the FRF as the impact force level is varied from approximately $95\text{N} \pm 5\text{N}$ to $1140\text{N} \pm 60\text{N}$ and show that the FRF does indeed change smoothly with changes in impact level. However, the effect of the changes seems to be quite substantial, and we surmise that it may be used to increase the effective energy absorption capability of interfaces.

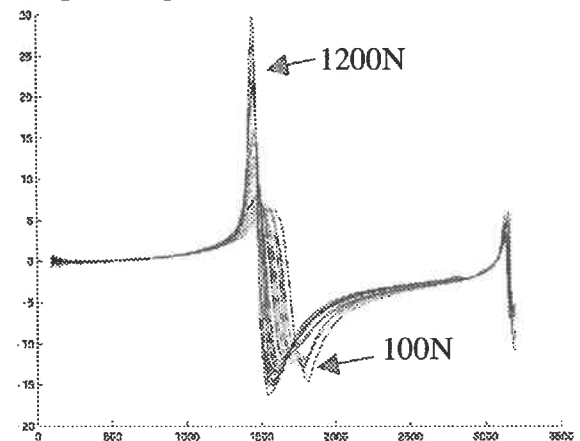


Figure 4 (a): Real part of FRFs with impact force level varied from $95\text{N} \pm 5\text{N}$ to $1140\text{N} \pm 60\text{N}$

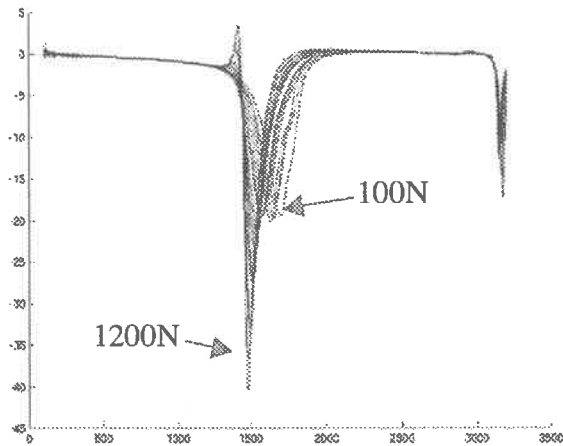


Figure 4 (b): Imaginary part of FRFs with impact force level varied from $95\text{N} \pm 5\text{N}$ to $1140\text{N} \pm 60\text{N}$

The mechanism for the apparent enhancement of energy loss is still not clear. Three possible mechanisms for losses in mechanical connections according to Rivin are material damping, external friction due to micro motions, and loss of energy in layers of lubricating fluid. According to Teng³, the distortion in the form of the FRF of the altered toolholder/spindle assembly suggests a softening spring. The previous work did not characterize the effect of hysteretic nonlinearities on the FRF. Since hysteresis does lead to a softening spring effect in addition to energy loss, we are conducting numerical simulations of hysteretic nonlinearities and comparing them to experimental results. As a preliminary model for the spindle-toolholder system, a simulation of the dynamic model of tangential compliance described by Rivin was developed in ABAQUS. In this model, a basic flat joint is examined with a thin deformable elastic strip in contact with a rigid base. The thin strip is subject to a uniform normal force and a cyclically varying axial load as shown in the figure below.

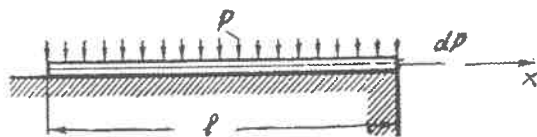


Figure 5: Model of a Clamped Connection from Rivin

The stress field at the interface between the two bodies resulting from contact friction during the first half cycle of the load (during ramp up to max load) was investigated with future work to include a cycling load. The figure below shows the expected stress field after the first half cycle of the cyclic axial load.

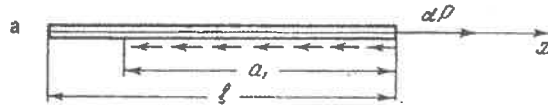


Figure 6: Model of interface stress field of a Clamped Connection from Rivin

Material type and load levels were chosen to match those seen in previous experiments with the tool holders. The results of the FE analysis are shown below. The deformation scale factor is 100.

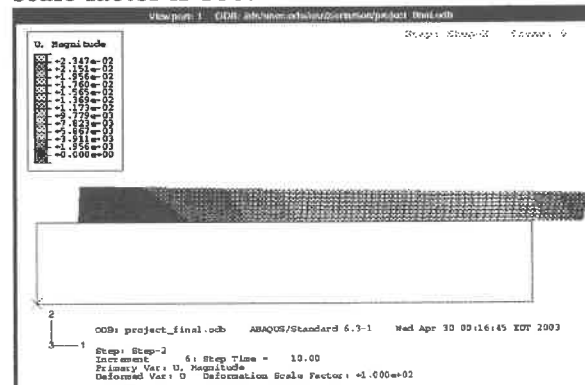


Figure 7: Results of tangential compliance FE analysis (plot of displacement magnitude)

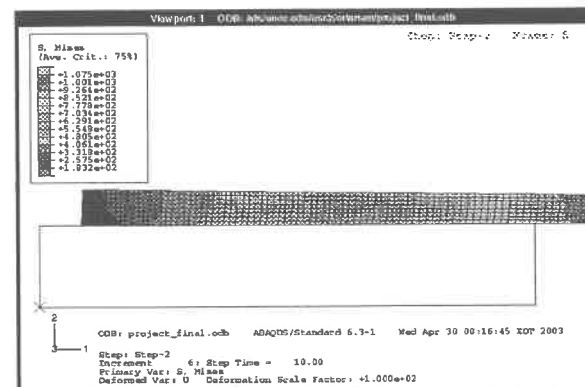


Figure 8: Results of tangential compliance FE analysis (plot of Mises stress)

Note how the deformed shape slopes towards the axial load due to Poisson's

effect and the motion of the bottom edge being retarded by the friction.

Also plotted are the friction shear stress (CSHEAR1) and the contact pressure at the surface nodes (CPRESS) at the final increment. If the value of $CPRESS * \mu > CSHEAR1$ at any given node, then it is sticking since the value of the shear stress is less than the critical value. If $CPRESS * \mu = CSHEAR1$ then the node is sliding. In Coulomb friction problems, the shear stress will never be greater than $CPRESS * \mu$.

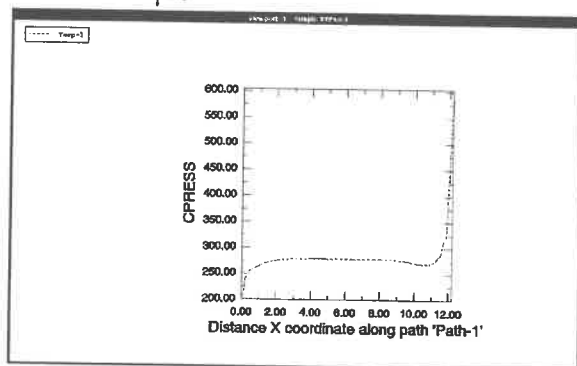


Figure 9: Plot of Contact Pressure

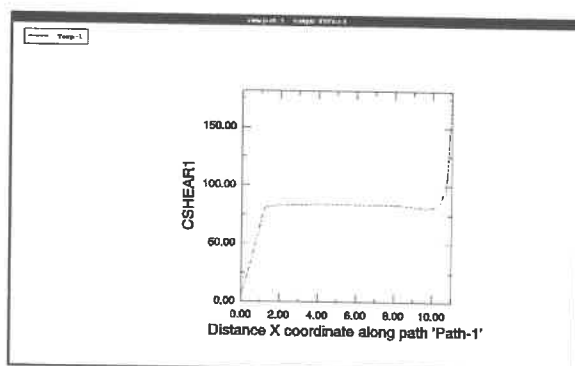


Figure 10: Plot of Contact Shear

The plot below illustrates the area of slip and stick of the interface.

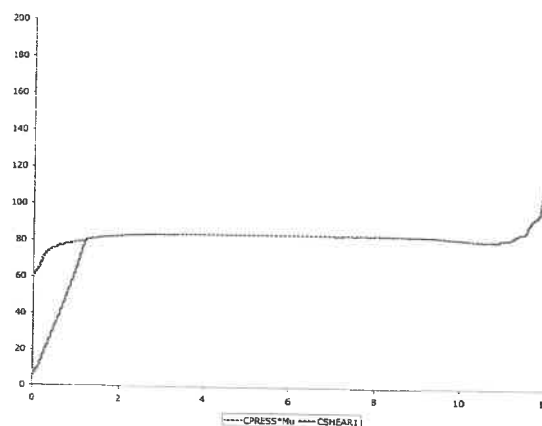


Figure 11: Plot of Slip Region

According to the above plot, nodes in the first 1.18 mm of the strip will stick with the applied 800 N load and the rest of the nodes to the right will slip. In addition, the near constant contact shear stress across the boundary once slip has occurred agrees with Rivin's analytical prediction. The spike at the right hand side is most likely a singularity associated with the application of the axial load on that face.

Acknowledgements

The authors would like to gratefully acknowledge financial support from the Caterpillar Corporation as well as helpful conversations with Brad Fix and Larry Carpenter of Caterpillar Corporation for this research.

¹ Jacobs, Characterization of the Machine Tool Spindle to Toolholder Connection, 1999.

² Rivin, Stiffness and Damping in Mechanical Design, 1999.

³ Teng, Simulation of the Impulse Response for Non-Linear System, 1986.

MACHINE FOR PRECISION HIGH-SPEED CUTTING OF SEMICONDUCTOR WAFERS

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Emerging applications in the microelectronics industry impose special manufacturing requirements that are not well addressed by conventional manufacturing techniques. Advances in laser technology, optics and beam steering combined with a better understanding of laser/material interaction make laser micromachining a viable, attractive, cost-effective and in some cases enabling technology to support these applications. Key applications include the following [1]:

- *Scribing low- κ dielectric materials*

The use of new low- κ dielectrics needed for performance enhancement brings numerous concerns, including thermally or mechanically-induced cracking or adhesion loss, poor mechanical strength, and moisture absorption. With dicing saws, large tensile and shear stresses are imparted at the cut zone, producing significant cracking and adhesion loss, leading to delamination of the metal and low- κ layers. The use of coolant during the sawing process leads to moisture absorption and material behavior which changes over time.

- *Dicing thin silicon wafers*

As the thickness of the silicon decreases, there is an increase in the vibration of the wafer during conventional sawing. This leads to problems with chipping and breaking of the dice. The increasing presence of metal for test structures on the wafer streets also impedes conventional sawing.

- *Dicing III-V semiconductors*

Compound semiconductors are starting to find increased application in RF and optical devices. The biggest application is in the light emitting diode market, where the substrates used are gallium arsenide, sapphire, gallium phosphide, and silicon carbide. In these applications, the cost of the substrate is high and the value of the wafers following epitaxy is even higher. However, due to the hexagonal symmetry, mechanical scribe and break leads to poor yield. Sawing uses up too much valuable real estate that could otherwise be used to make real devices.

The ESI Model 2700 Micromachining System uses a laser to cut semiconductor wafers. By focusing the laser beam to a small spot size, the machine is able to cut features of any shape with a cut width as small as 10 μm while maintaining a cutting speed that rivals that of saws for many applications and keeping beam positioning errors under 10 μm over the entire area of a 300 mm wafer. Two different mechanisms are used to steer the laser beam over the surface of a wafer. Large motions of up to several hundred millimeters are made using translation stages mounted on recirculating bearings and powered by linear motors. One stage carries the wafer, while a perpendicular stage carries optical elements to position the laser beam. Small beam positioning motions are performed by a pair of galvanometric scanners mounted on the optics stage, which deflect the beam through a telecentric scan lens and onto the wafer, in a 2 mm square field. A drawing of the laser beam positioning system is shown in Figure 1.

Most traditional laser machining systems use step-and-repeat positioning. They use stages to statically position the wafer, while the scanners direct the beam to any point inside a step-and-repeat field. This can lead to two problems. First, it requires large step-and-repeat fields to minimize the number of slow stage moves between fields. This has the effect of increasing both the cost of the scan lens and the nonlinear positioning errors at the edges of the scan field. Second, it could lead to abutment errors of features crossing the edges of step-and-repeat fields.

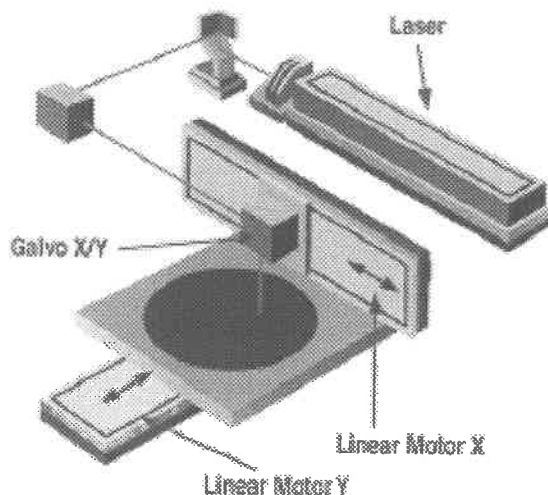


Figure 1. The laser beam positioning system used on the ESI Model 2700 Micromachining System. The beam exits the laser, is deflected by several stationary turn mirrors, is deflected downward by a mirror which moves on the X stage, is directed in X and Y by the galvanometric scanners, then hits the wafer being carried on the Y stage.

The ESI machine avoids the problems inherent in step-and-repeat beam positioning mechanisms by using a patented compound beam positioner [2,3]. The large stages move continuously, tracing the bulk of the beam's desired motion across the wafer, while the scanners steer the beam to generate the fine details of the cut and compensate for any large-stage positioning error. Cuts are made continuously, avoiding abutment errors, while motion between cuts is almost always made by the fast scanners, rather than the slower large stages. The compound beam steering mechanism can thus cut profiles faster and more accurately than a traditional machine. The size of the scan field can be optimized based on speed, accuracy and cost requirements.

A diagram showing the method used to synchronize and control the motions of the stages and scanners and the pulsing of the laser is shown in Figure 2. The laser beam

motions required to process a given wafer are calculated before processing begins. All moves are profiled with sinusoidal acceleration, subject to a preset minimum period for the acceleration/deceleration segments of moves and maximum acceleration and velocity limits. The calculated position and acceleration profiles are passed to a fourth-order low-pass filter and then on to the stage controller, where the acceleration is used for feedforward and the position is used for feedback control.

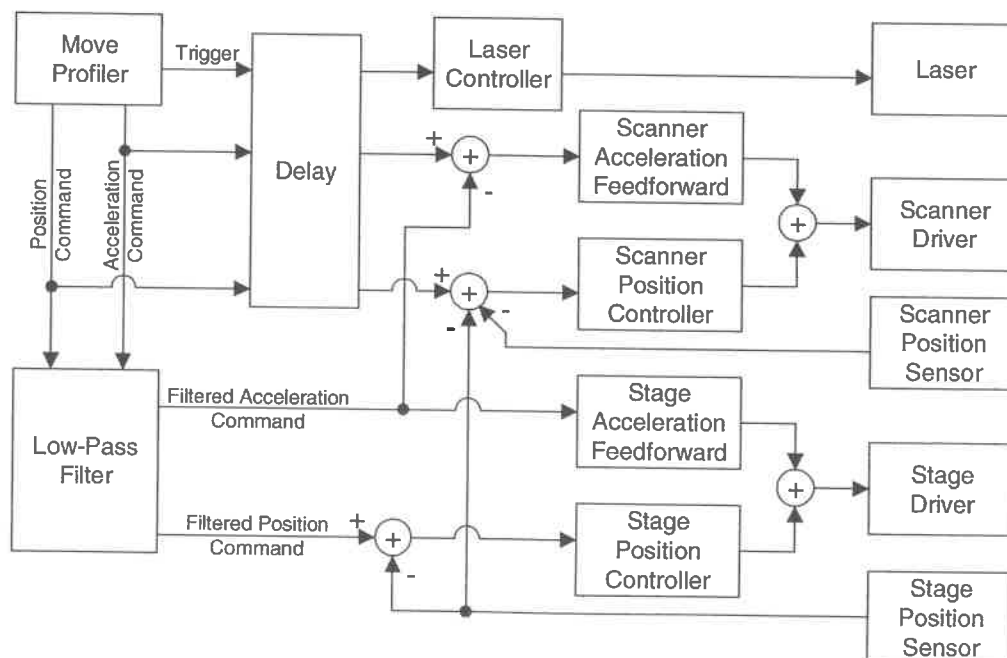


Figure 2. Diagram of method used to control the coordinated motions of the stages and scanners.

The low-pass filter produces phase lag in the stage control signal. If the delay was allowed to remain in the control loop, the scanners would constantly be moving toward their ends of travel as they move the laser beam toward the next hole. To keep the laser beam toward the center of the scan field, a delay is added to the calculated position and acceleration profiles, and these delayed commands are fed to the scanner controller. The scanner controller subtracts the low-pass-filtered stage commands, so the resulting command contains just the higher-frequency components of the total positioning command. As mentioned above, the stage position error (from the filtered stage command) is also fed back into the scanner position command, so the scanners can quickly compensate for whatever stage errors are present, while the stage's controller corrects the errors more slowly.

The ESI Model 2700 also uses special calibration methods to reduce beam positioning errors even further. Using a small amount of scanner angular travel eliminates the need for non-linear calibration of the scan field, minimizing barrel and pincushion optical distortions. Linear stage static errors are then corrected by mapping the workspace with a high-accuracy calibration grid using a machine vision system mounted parallel to the scan lens. Typical uncorrected and corrected errors of the linear stages are illustrated in Figure 3. Application of software correction reduces the static error across the part from 17 μm down to 2 μm . Next, the laser to camera offset, measured as a function of stage travel to account for varying Abbé errors, is corrected by cutting a sequence of features on a wafer and measuring them with the vision system.

Following calibration, during run time, when the wafer is loaded on the machine, the machine vision system looks at alignment features on the wafer and applies coordinate transformations to the toolpath on top of the pre-calculated error corrections. Finally, while the wafer is being cut, the high-bandwidth scanner controller monitors linear stage dynamic position error and moves the scanning beam to compensate. Thus the machine achieves total beam positioning error of less than 8 μm over the entire area of a 300 mm wafer with continuous motion and without stitching step-and-repeat fields. To illustrate this, holes were drilled on a silicon wafer and referred back to the alignment grid using a Nikon

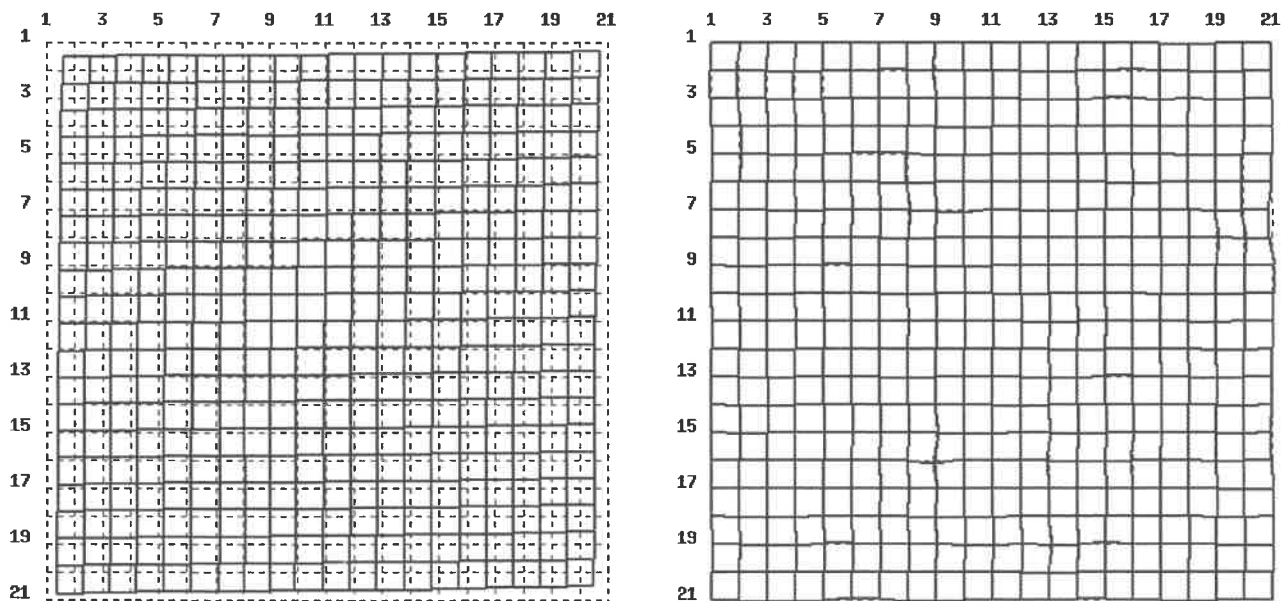


Figure 3: Uncorrected and corrected error plots over a calibration grid. The plot on the left shows uncorrected position, with a maximum error of 17 μm , while the plot on the right shows corrected position with a maximum error of 2 μm .

coordinate measuring machine. A plot of the registration errors of the holes is illustrated in Figure 4. The graphs show that the corrected errors are kept to $\pm 2.5 \mu\text{m}$ in this test. The normality of the error histogram further illustrates that almost all repeatable geometric errors have been removed using software correction.

The novel beam positioning method of the ESI Model 2700, combined with its thorough error compensation and high-power laser, enable semiconductor manufacturers to maximize their usage of the available space on a wafer and achieve significant quality and throughput gains over traditional dicing machines.

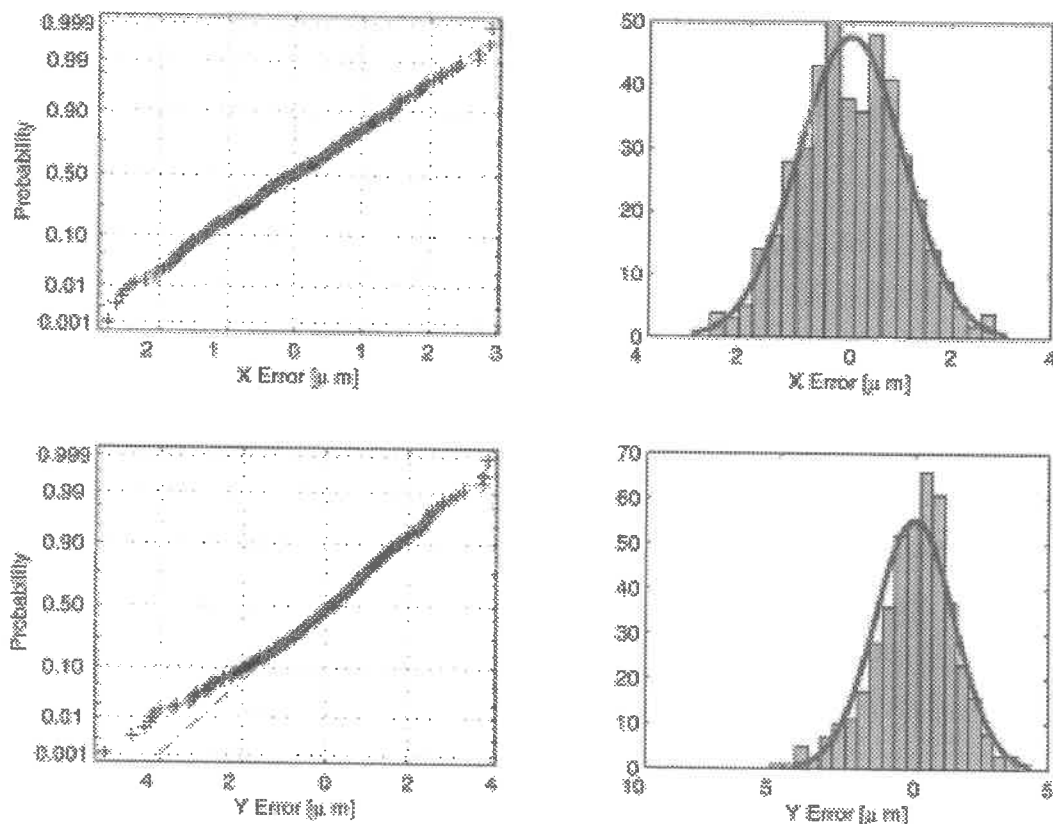


Figure 4: Registration errors, illustrating complete removal of repeatable geometric errors. The lower pair of graphs shows uncorrected error, while the upper pair of graphs shows the error after correction.

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Traceable calibration of non-linearities in laser interferometers

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ABSTRACT

Laser interferometer systems are known for their high resolution, and especially for their high range/resolution ratio. In dimensional metrology laboratories, laser interferometers are popular workhorses for the calibration of measurement equipment. The uncertainty is usually limited to about 10 nm due to polarization- and frequency mixing. For demanding applications however nanometer uncertainty is desired. We present the measurements of interferometer components, which will be used to calculate an integral non-linearity by a virtual interferometer, of which the results will be verified with a traceable Fabry-Perot interferometer with sub-nm uncertainty.

Keywords: Laser interferometry, heterodyne, sub-nm, non-linearity

1. INTRODUCTION

The fundamental limitations of laser interferometer systems for achieving (sub-)nm uncertainties in small displacement measurements are in the photonic noise and in residual non-linearities which are inherent of periodic interferometer signals and which repeat each fraction of a wavelength. These deviations result from both the laser and from the phase measurement system of the interferometer itself,¹ but they can also result from polarization states of the laser and the optics used: beam splitters, retardation plates used with plane mirrors and corner cubes can influence the polarization state in interferometers using polarizing optics, or influence the contrast in interferometers with non-polarizing optics. We work with heterodyne laser interferometers, which use a polarizing beam splitter.

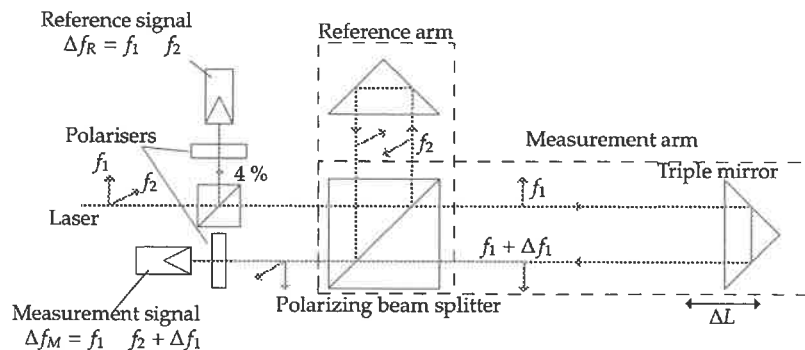


Figure 1. Schematical representation of the principle of a heterodyne laser interferometer.

In figure 1 the schematical principle of a heterodyne interferometer is presented. As a result of non-ideal optics and laseroperation the two beams emerging from the laser source will not be orthogonally linear polarized. Instead, they will be elliptically polarized² and non-orthogonal. Further the optics used, inherit polarizing properties themselves resulting in non-ideally split polarizations. Finally the effect of misalignment in optics must be mentioned. All these effects result in polarization mixing in the measurement and reference beams. As a result non-linearities exist in the interferometer measurements. Non-linearities with a periodicity of one wavelength of optical displacement are called first order non-linearities.

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2. SIMULATED NON-LINEARITY OF A COMMERCIAL LASERINTERFEROMETER

Based on a Jones-matrix formalism a model was build enabling the calculation of non-linearities out of different sources of non-linearities as mentioned before. The advantages of a Jones matrix formalism above an analytical formalism³ is the possibility to include the polarization properties of optics and the simplicity to change the optical setup of the model. Further, by implementing a Fourier analysis to the model a clear representation of first and second order non-linearities is gained. As an example the non-linearity was calculated resulting from an ideal lasersource with rotated ideal flat-mirror optics, the results are shown in figure 2(a).

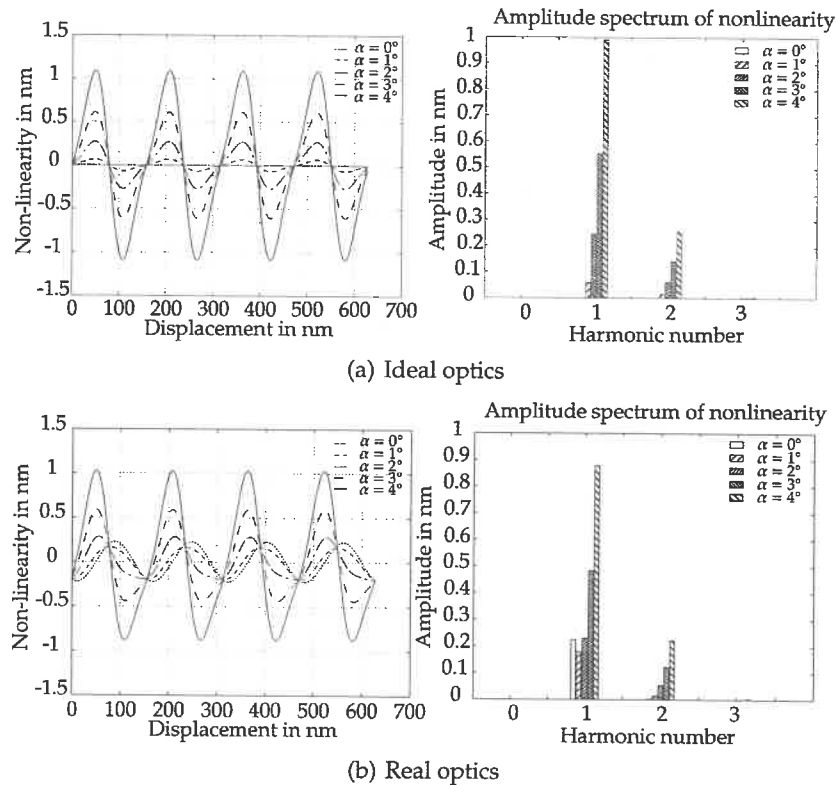


Figure 2. Influence from polarization properties of interferometer optics on calculated non linearity for rotated optics with angles (α) varying from 0° to 4° .

With use of an ellipsometer the polarizing properties of those flat mirror optics were investigated. The setup consisted on a null-measurement and a four-zone method.³ The latter to exclude effects of the ellipsometer optics. The results were: A leakage of 3.2% for the polarizing beamsplitter and a surprising quarter waveplate phase error between fast and slow axis of 7.85° . Further a leakage of less than 0.2% was measured between the waveplate transmission axes. The ellipticity of the laserhead was measured using a circular polarized laser source, a polarizer and a spectrum analyser, as described by Lorier.² The ellipticity was 1 : 148 in E-field for the first frequency and 1 : 164 in E-field for the second frequency. Substituting these numbers into the model results in a calculated non-linearity as shown in figure 2(b). As can be seen in the right side of figure 2(a) the result of an ideal interferometer with rotated optics is a first order non-linearity with a nearly linear dependence of the rotation angle in combination with a second order non-linearity. By implementing the polarization properties of the interferometer optics in the model this linear dependency is disturbed as shown in the right side of figure 2(b). Thereby also disturbing the symmetry in the non-linearity, due to a relative larger second order non-linearity as can be seen in the left side of figure 2(b).

3. TRACEABLE CALIBRATION SETUP

In the section Precision Engineering of the Eindhoven University of Technology a calibration facility was developed with a range of $300\text{ }\mu\text{m}$ and an uncertainty of 1.2 nm .⁴ Since then the HeNe slave laser was replaced by a narrow linewidth diode laser, see figure 3(b). The measurement principle is based on a Fabry-Perot cavity to which a diodelaser is frequency locked. When one of the mirrors of the Fabry-Perot interferometer is displaced, the slave laser tracks the change in resonance frequency of the cavity. At the same time the displacement of the upper mirror is measured by a commercial displacement laser interferometer system, an Agilent 5528 system (see figure 3(a)). The frequency counter measures the frequency difference between the slave laser and an iodine stabilized HeNe-laser. A comparison of the laser readout with the displacement derived from the frequency measurement gives the calibration data.

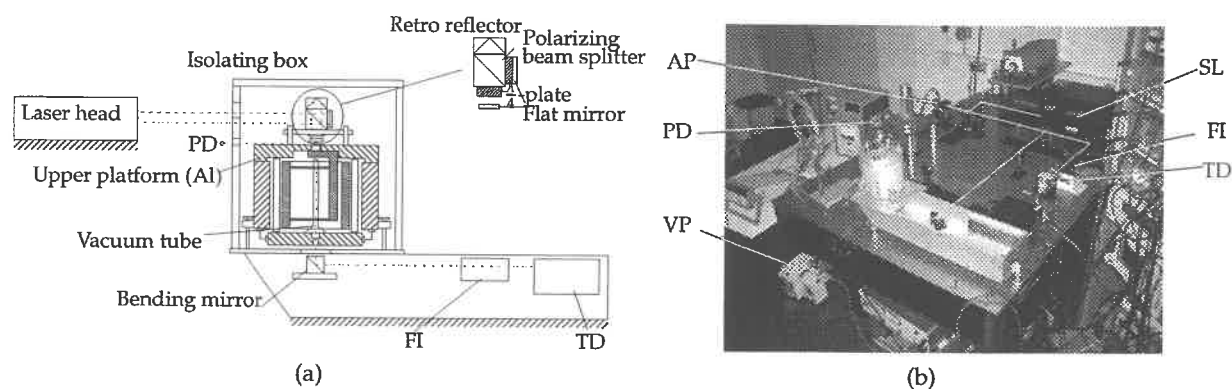


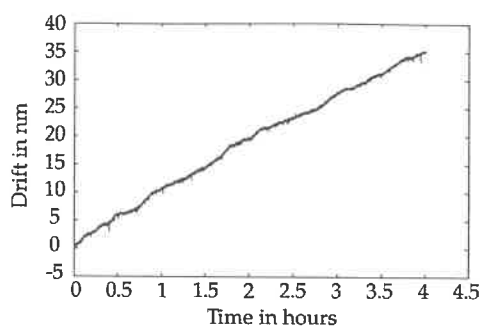
Figure 3. The used Fabry Perot calibration setup with SL: Stabilized HeNe-laser, TD: tunable diode laser, FI: Faraday Isolator, PD: Photo diode, VP: Vacuum pump, AP: Avalanche photo diode.

3.1. Measurement results

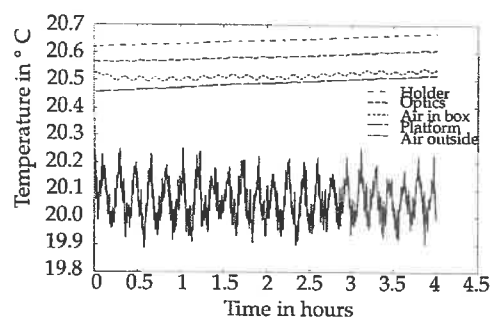
First the drift of the setup was determined. This drift is shown in figure 4(a). The drift was 35 nm in 4 hours during a linear temperature change of 0.045° of the holder and optics. After careful analysis it seems most likely that this is the effect of thermal expansion of the optics itself in combination with a change in refractive index. A calibration was carried out with an Agilent 5528 laser system combined with the flat mirror optics, of which the polarization properties were measured earlier with ellipsometry. The schematics of the setup, together with a photograph are shown in figure 3(b). In this photo the isolating box was removed together with the thermistors. Since this Agilent laser cannot be triggered extensive averaging must be carried out. With the shown drift a compromise must be made between the measurement range and accuracy. The first calibration was carried out over a range of 650 nm back and forth with a rotated laser head (3.3°), due to a non-horizontal platform the rotation seems exaggerated in the photo. The non-linearity was modelled also and the results are shown in figure 5(a). The asymmetry mentioned in section 2 can be seen also in this measurement. As can also be seen the measurement noise is quite large as a result of fast measurement. The standard deviation of the model compared to the measurement was 0.3 nm . In order to test the non-linearity with a well-aligned laser head, a range of 180 nm was scanned back and forth with large averaging. This measurement is shown in figure 5(b) together with the simulation. In this case the standard deviation of the model compared to the measurement was 0.14 nm .

4. CONCLUSIONS

Presented are the results of the modelling of non-linearity in a commercial heterodyne interferometer with use of the measured polarization effects of laser source and optics. The model was verified with a traceable calibration on a Fabry-Perot cavity. From this it can be concluded that the effect of polarization properties on the non-linearity of the laser interferometer is significant, and should be considered when making measurements with nm accuracy.

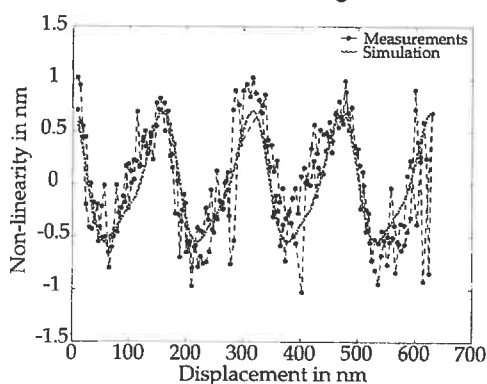


(a) Drift of setup

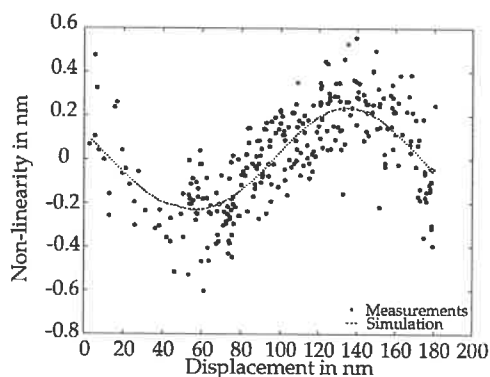


(b) Temperature

Figure 4. Measurements with the Fabry-Perot setup



(a) Optics rotated 3.3°



(b) Well aligned

Figure 5. Measurements with the Fabry-Perot setup

Acknowledgement

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INVESTIGATION OF OPTICAL CLEAN ZONE BOUNDARIES USING PHOENICS

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Introduction

In machining processes, coolants are used to reduce the effects of excessive thermal energy existed due to the interactions between workpiece and cutting tool. However, the opaque optical property of the coolants causes an inaccessibility problem for in-process optical measurement of the workpiece surface profile. To solve the problem, a transparent fluid beam is ejected from a beam applicator to make an optical clean zone on the workpiece surface and the laser beam of a triangulation sensor can pass through that area to access the surface. For effective measurement, the size of the clean zone should be established and should be sufficiently large. If the region is too small, the measurement range of a laser sensor cannot be fully utilized. In some cases, the measurement cannot be achieved at all. The formation of the clean zone is affected by several factors including the velocity of the coolant, the velocity of the transparent fluid, the velocity of the workpiece surface, and the gap between the workpiece surface and the transparent fluid beam applicator. In order to investigate the flow patterns for different variables, an experimental system with a coolant and fluid injection system, a flow visualization system, and a triangulation sensor was constructed. However, the testing had a limitation that it will be difficult to perform tests in a large number of configurations. In the meantime, the solutions based on theoretical models for the complex interactions between the coolant and the clean beam may be difficult to obtain [1-5]. To address this issue, a widely recognized computational fluid dynamics CFD package Phoenix was used. For higher fidelity of the findings, and to find the clean zone forms, further experimental and computational tests were carried out. For the objectives, the Phoenix package was used to investigate the parameters affecting the construction of the optical clean zone. One particular emphasis of the investigation is to obtain data to establish models to describe the boundaries of the clean zone. The new findings can be validated by the experimental results obtained from the visualization system and an image processing algorithm in our previous study, demonstrating the effectiveness of the computational method in obtaining the boundaries of the optical clean zone.

System for In-process Measurement Testing

The fluid system is modeled as a transparent fluid ejected from a hole of an injection applicator which is located in the unidirectional flow of coolant fluid and at the top of workpiece surface with a gap g (Fig. 1-2). For simplification, the transparent fluid beam and the coolant are assumed to be Newtonian fluids, isothermal, and incompressible. The coolant flow is assumed to be laminar and at a steady state. The fluid beam flow is assumed to be uniform and at a steady state. The experimental setup and the image of the optical clean zone, the opaque area, are shown in Fig. 2-3. Different boundary conditions were considered. The values of the velocity of the transparent fluid v_t , the velocity of the coolant v_c , the velocity of the lower plane v_{bottom} that is the velocity of the workpiece surface, the velocity of the upper plane v_{top} that is the velocity of the

surface of the injection applicator, and the distance of gap g are described in Fig. 1 and Fig. 4-12. Equations to model the system are based on the continuity equation and the Navier-Stokes equations. It was discussed in our study performed previously [1]. Phoenixics is recommended for the computational testing because it can provide a clear visualization of velocity distribution in a small region.

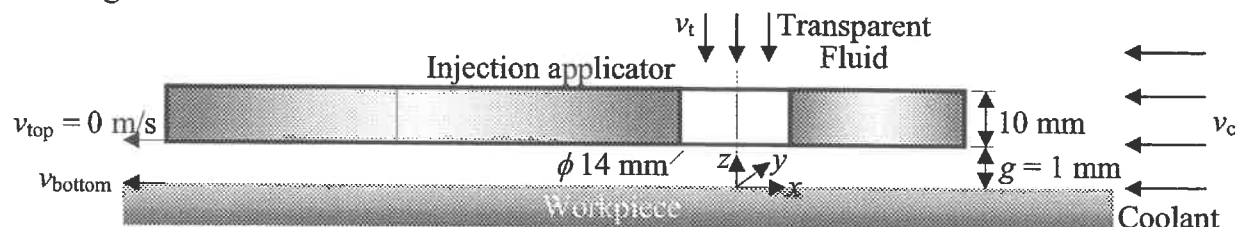


Fig. 1 Coolant and transparent fluid injection system



Fig. 2 Experimental setup

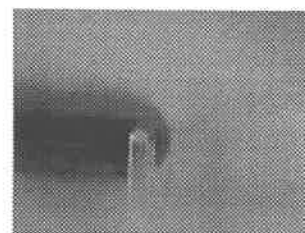


Fig. 3 Optical clean zone

Computational Testing and Discussion

Figure 4 shows a Phoenixics result of a source of transparent fluid beam in the x - y plane. It shows a radial flow and the velocity is decreasing from the centre to the outwards. Figure 5 provides the result of a laminar coolant flow between two fixed parallel planes in the x - z plane. The result shows that the velocity distribution is symmetric along the centre plane of the parallel planes and the flow is fully developed. The maximum velocity is at the centre plane and the least velocity is near the surfaces. It is because there are shear stresses exerted on the coolant by the surfaces of the injection applicator and the workpiece. Figures 6-8 give the laminar flow between the planes with one fixed and one moving. The velocities of the lower plane v_{bottom} , that is the velocities of workpiece surface, are 0.3, 0.5 and 0.8 m/s, respectively. The symmetric property is not hold. If the velocity of the lower plane is high, the maximum velocity is closer to the lower surface. The complex velocity for the flow of the coolant and transparent fluid injection system is the sum of the velocities for the source and the laminar flow between the parallel planes with one of the surface being fixed and the other one moving. Three velocities of the transparent fluid, $v_t = 0.1$, 0.18 and 0.28 m/s, are shown in Fig. 9-11, respectively. The results show that if the velocity of the transparent beam is increased, the influence to the coolant is enhanced. Fig. 12 is an enlarged image of Fig. 11. Straight lines are drawn from the center of the transparent fluid beam at different angles in the x - y plane. The velocity distribution from the centre to the outward position along the lines can be divided into several segments based on the value of the velocity. The length of the segment is decreased from the centre to a certain point and then increased from this point to the outer location. This particular point will form a point of the optical clean zone boundary. The decrease in the distance is because the velocity of the outflow of the transparent fluid source from the centre is reduced when the travelling distance is increased. Beyond the boundary point, the distance is increasing, as the influence of the source is reduce and the coolant

flow plays a more important role. It should be noted that the velocity of the coolant is higher than that of the transparent fluid in the defined boundary condition (Fig. 9-11). Several points are marked in Fig. 12 and the boundary can be found using a curve passing through all the points. A technique to process the experimental image of the optical clean zone [2] was used. The processed image for Fig. 3 is shown in Fig. 13. It shows that the computational boundary shown in Fig. 12 is similar to the experimental boundary shown in Fig. 13. Several straight lines are drawn in the same way as before. The marked points (Fig. 13) are the results of intersection of the straight lines and the boundary. The distances between the boundary point and the centre in the computational testing and the experimental results are compared. The average error for the examined cases (Fig. 9-11) is better than 15%. The error is acceptable because the measurement area for the triangulation sensor will be set 10-15% smaller than the optical clean zone such that disturbances near the boundary could not affect an in-process optical measurement.

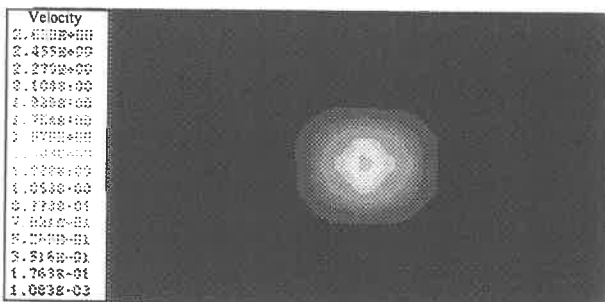


Fig. 4 Velocity distribution for $v_c = 0$ m/s, $v_t = 0.8$ m/s, and $v_{\text{bottom}} = 0$ m/s

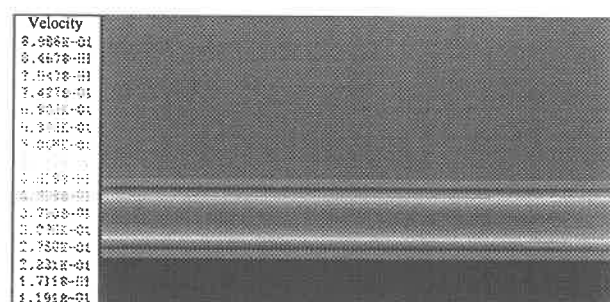


Fig. 5 Velocity distribution for $v_c = 0.6$ m/s, $v_t = 0$ m/s, and $v_{\text{bottom}} = 0$ m/s

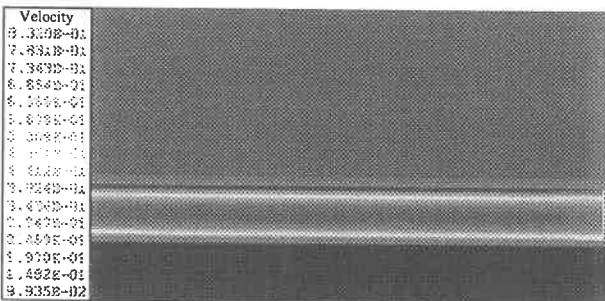


Fig. 6 Velocity distribution for $v_c = 0.6$ m/s, $v_t = 0$ m/s, and $v_{\text{bottom}} = 0.3$ m/s

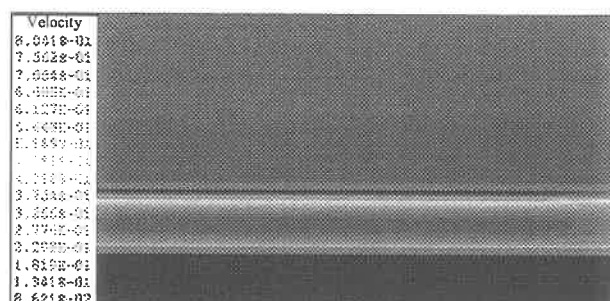


Fig. 7 Velocity distribution for $v_c = 0.6$ m/s, $v_t = 0$ m/s, and $v_{\text{bottom}} = 0.5$ m/s

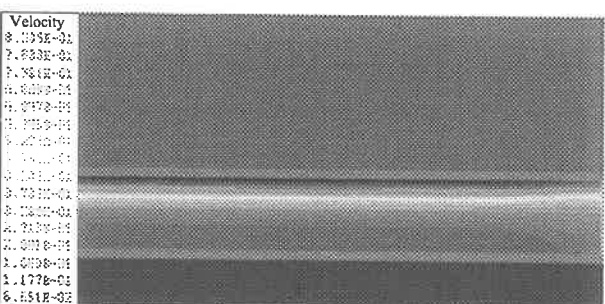


Fig. 8 Velocity distribution for $v_c = 0.6$ m/s, $v_t = 0$ m/s, and $v_{\text{bottom}} = 0.8$ m/s

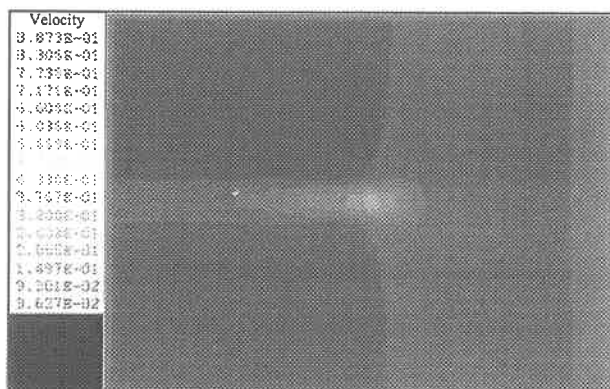


Fig. 9 Velocity distribution for $v_c = 0.6$ m/s, $v_t = 0.1$ m/s, and $v_{\text{bottom}} = 0.3$ m/s

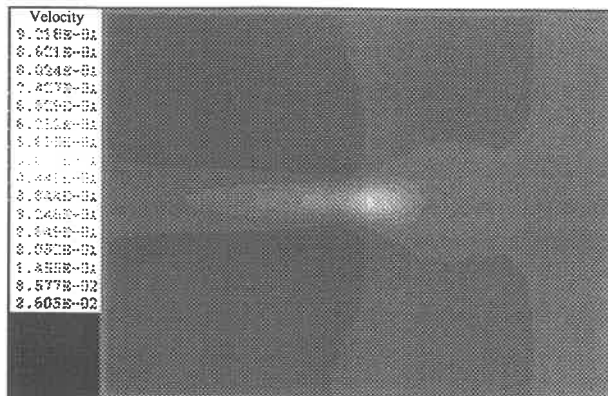


Fig. 10 Velocity distribution for $v_c = 0.6$ m/s, $v_t = 0.18$ m/s, and $v_{\text{bottom}} = 0.3$ m/s

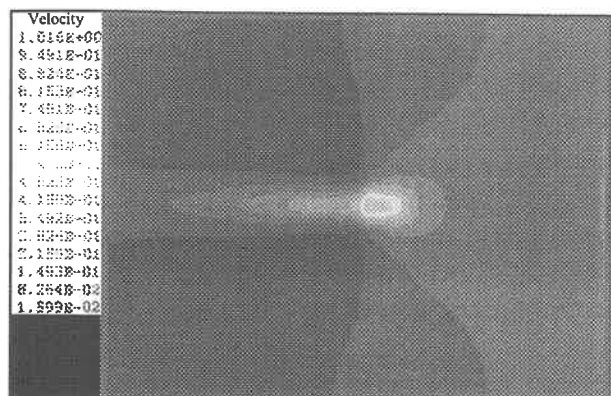


Fig. 11 Velocity distribution for $v_c = 0.6$ m/s, $v_t = 0.28$ m/s, and $v_{\text{bottom}} = 0.3$ m/s

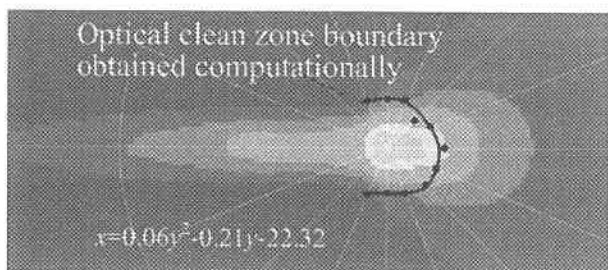


Fig. 12 Velocity distribution for $v_c = 0.6$ m/s, $v_t = 0.28$ m/s, and $v_{\text{bottom}} = 0.3$ m/s

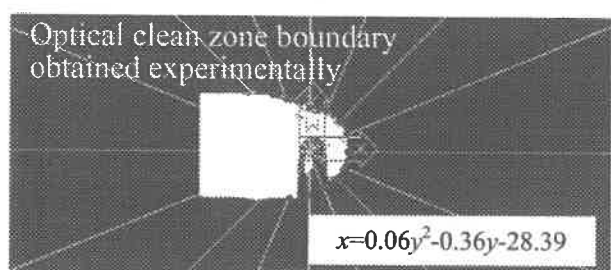


Fig. 13 Velocity distribution for $v_c = 0.6$ m/s, $v_t = 0.28$ m/s, and $v_{\text{bottom}} = 0.3$ m/s

Conclusions

A computational method in obtaining boundaries of the optical clean zone is presented. The computational results for different boundary conditions are obtained using the proposed method. The results can be confirmed by the experimental results. The method allows a series of computational tests be carried out for different configurations and the need for a large number of the experimental tests is avoided.

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Design and Analysis of Shrink and Press Fit Joints

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Introduction

Pulsed Power experiments under way at Los Alamos National Labs use short pulses ($<5 \mu\text{sec}$) of high current ($>10^7$ amps) to create a large magnetic force on the cylindrical "liner" carrying this current. As a result of this force, the liner implodes onto a target positioned at its centerline. Creating a large force on the target is the goal of the experiments. The liner is a thin-walled aluminum cylinder (thickness of 1 mm with a radius of 50 mm) captured at the ends by two copper guide planes using either press or shrink fits. The contact between the liner and glide planes is critical to the success of the experiment. The analytical challenge for investigating these components is predicting the presence of a partial shrink fit; that is, a shrink fit whose length is less than the apparent length of the interface between mating cylinders. Such stress and deformation distributions are difficult to calculate using analytical shrink fit theory because of the varying radial deflection profile along the length of the liner. A second goal is to design an initial non-cylindrical liner such that when it is assembled onto the glide planes, it will assume a cylindrical shape. This cylindrical geometry will lead to uniform implosions of the liner and are deemed to be highly beneficial for the success of the experiments.

A model of the liner/glide plane interface was created using the finite element code, LS-DYNA, distributed by Livermore Software Technology. This code is a general-purpose, transient dynamics finite element program originally developed for simulations of projectile penetration, blast response and explosives. It has the ability to model inelastic material behavior and to treat contact constraints. The FE model was used to address the following: mechanics of the interference fits, physical description of the contact surfaces between the liner and glide planes, joint void (gap) description, shape of the assembled liner and material property effects. Different shrink fit geometries were investigated to evaluate their effect on the assembled shape, joint interface gaps and low interface stress magnitudes. The results of the finite element models were also compared to a scale-model of the liner/glide plane system built at the PEC. The radial deflection of the diamond turned thin-walled liners fitted to glide planes were measured using an optical interferometer as well as a contact profilometer. These measurements verified the FEM results and the technique proved to be a tremendous aid in accurately predicting the shape, deflection, and stress distribution of such composite cylindrical elements.

Model and Results

Copper glide planes are inserted in either end of the cylindrical liner, typically via a thermal shrink fit process, though more recent designs have used a modified geometry with a mechanical press fit process. These glide planes serve as a mechanism to transfer tremendous electrical currents to the liner, which leads to a cylindrical liner implosion. To optimize the results of these experiments, it is necessary for this cylindrical implosion to occur very uniformly. Therefore, manufacturing dimensionally precise liners with minimal surface flaws is essential. Unfortunately, the shrink fit process naturally results in residual stress and distortion in the liner

walls, which need to be compensated. A schematic illustrating the assembly orientation of a single wall liner with glide planes is shown in Figure 1. The nominal diameter of the liner is 100mm, the height is 55mm, the wall thickness is 1.2953mm, and the nominal interference fit is 15.24 μ m (0.0006in).

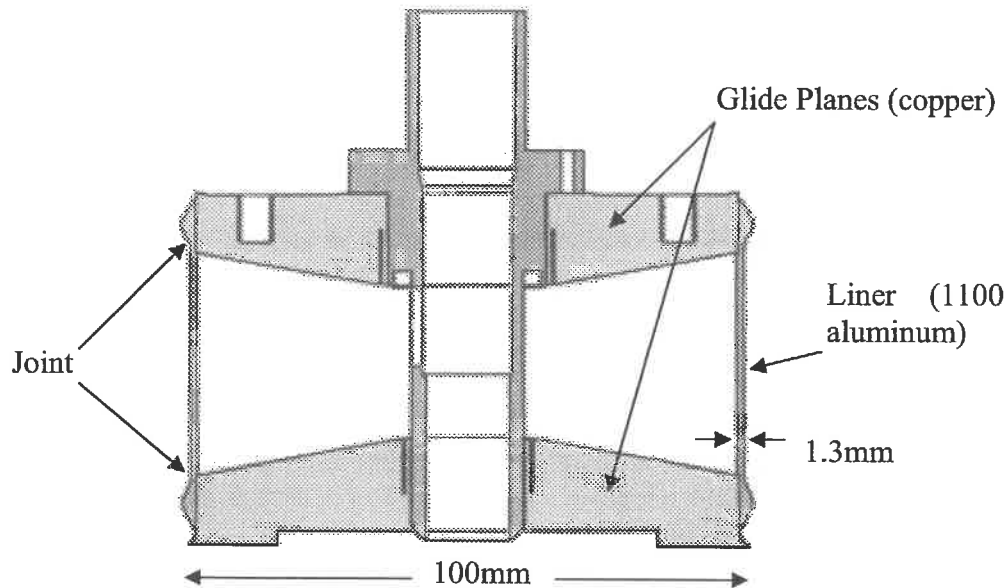


Figure 1- Cross section of liner and glide planes (NTLX Shiva Star/Atlas Configuration)

Figure 2 shows the baseline mesh modeling the geometry of this axi-symmetric system. The axi-symmetric modeling does ignore some details of the glide planes, such as holes and ports, but this is not expected to impact the results of interest in any significant way. Each component is modeled in LS-DYNA as an elastic-plastic material, thus any yielding that could occur is considered. The liner is composed of 1638 elements with 8 elements through the thickness, and the large and small glide planes are composed of approximately 2000 elements each. For this preliminary FEA model in particular, a composite liner is *not* used. A uniform temperature change of -100°C was imposed on the glide planes during simulation. The parts were then slipped together, followed by a return to ambient temperature with contact modeling between the parts activated during the transition to equilibrium. The glide planes expanded during this phase, deforming the liner as expected.

A displacement vector plot for nodes within the liner is shown below in Figure 3a and 3b. The lengths of the arrows indicate magnitude of radial displacement in the liner wall, with the peak deflections occurring at either end of the liner. Note that the vectors are magnified for an enhanced visual representation of the displacement behavior. The displacement decays with distance from the liner ends and actually becomes negative, indicating inward radial movement at the mid-length position of the liner.

Figure 4 shows data along the longitudinal axis (3 data scans 120 deg. apart) on the OD of a scale-model liner/glide test specimen in comparison to the LS-DYNA liner deflection profile. This data was obtained with the TalySurf Profilometer. Excellent agreement is apparent in this comparison, providing a validation of the LS-DYNA contact modeling capability.

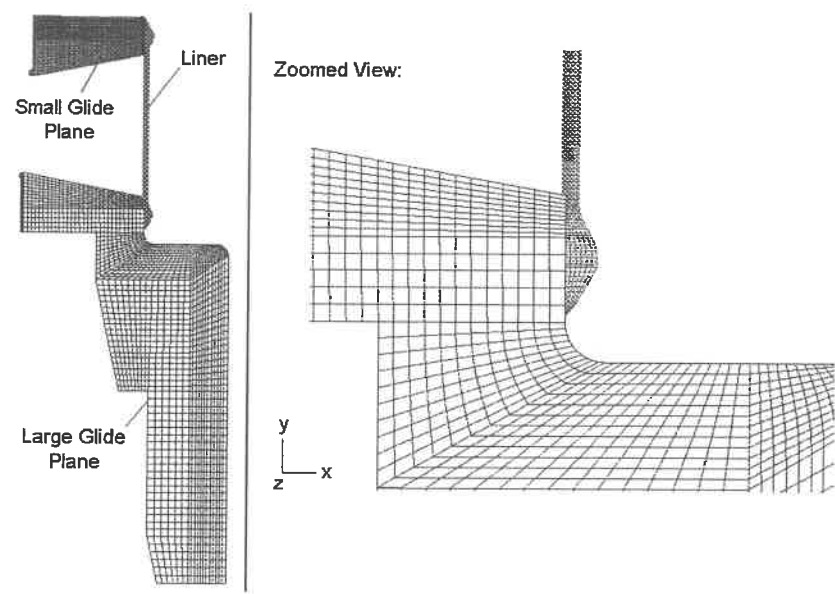


Figure 2- LS-DYNA mesh used for LANL prototype simulation

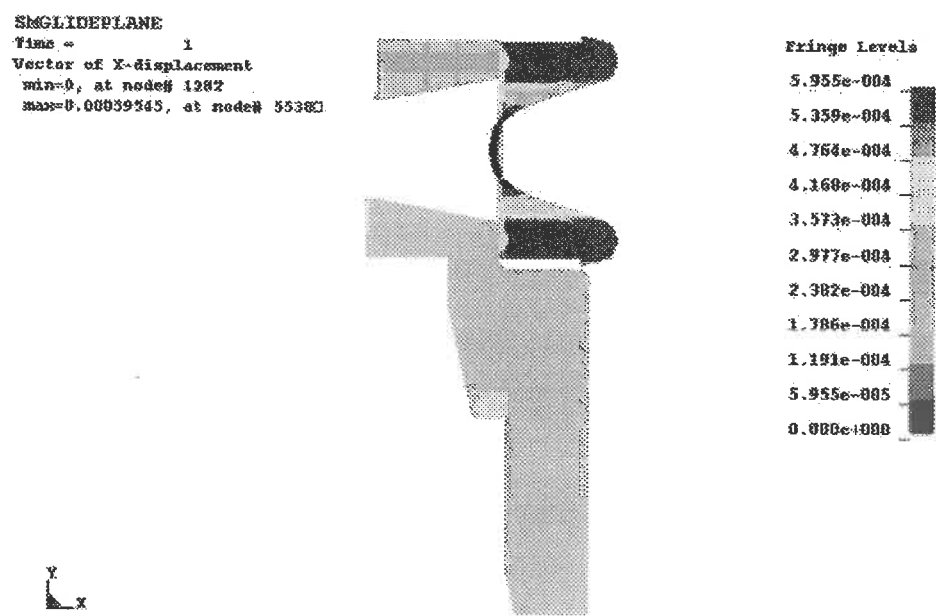


Figure 3a- Magnified vector plot of radial displacement along the liner

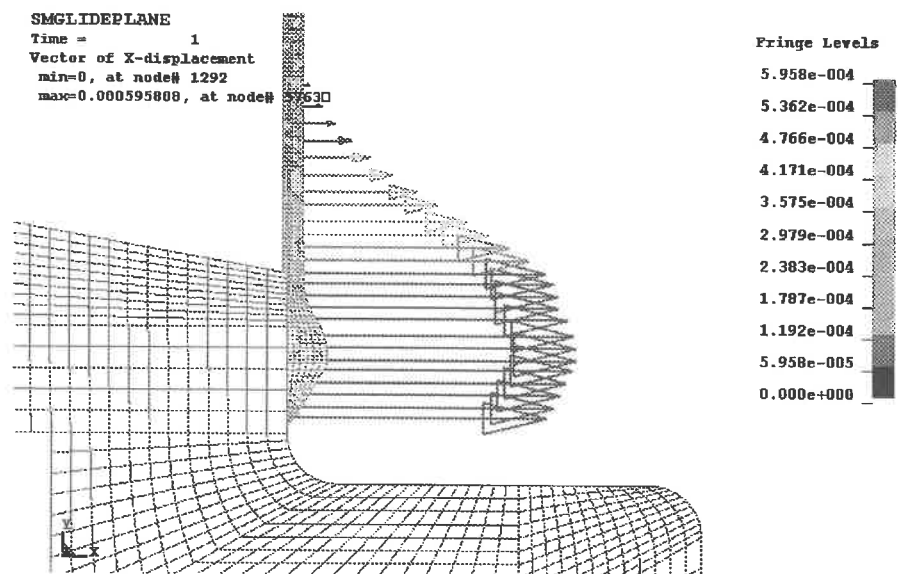


Figure 3b- Zoomed view of the radial displacement vector plot

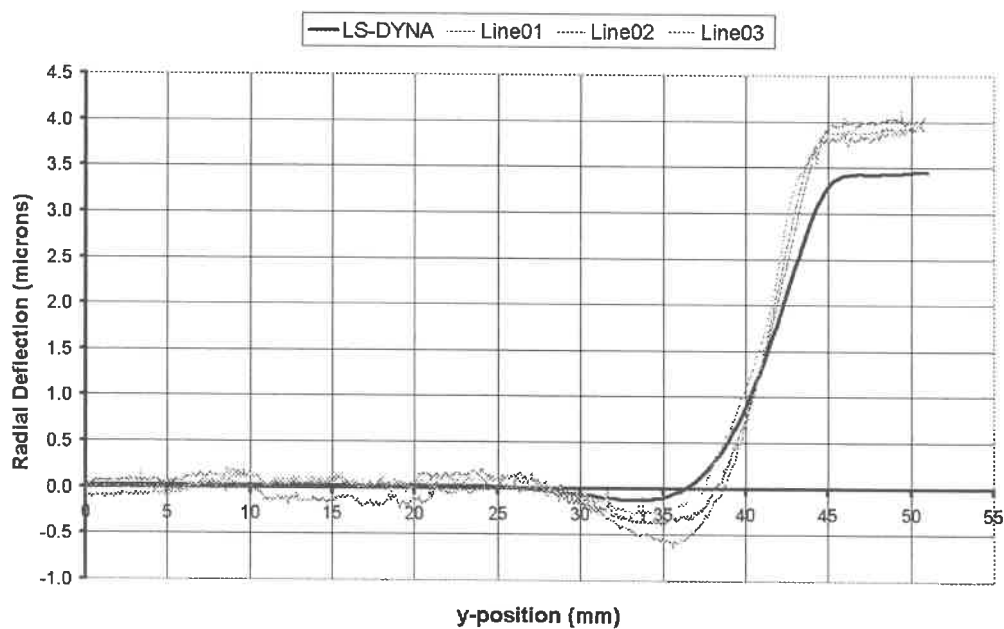


Figure 4- Comparison of LS-DYNA and scale-model test specimen liner displacement values for three longitudinal scans

Time Series Modelling of Endmilling Process for Mode Estimation

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ABSTRACT

To get the natural mode of structural dynamics considering cutting process dynamics it is necessary to model the cutting system. Several time series modelling such as AR (burg, least square, yule walker, geometric lattice, instrument variable) algorithm, ARX (arx, iv4), ARMAX, ARMA, Box Jenkins, Output Error, recursive arx and recursive armax were analysed and compared with one another. As a result, arx, armax and iv4 were proved very reliable algorithms for the calculation of the modal parameters(natural frequency and damping ratio) in endmilling operation.

INSTRUCTION

In this study, we have modelled several time series algorithms with spectral analysis comparing with conventional FFT (fast fourier transform) method and calculated the natural mode with each algorithm in endmilling operation. The natural mode estimations with several time series algorithms may be practiced for latter study of chatter. Also these results may be compared with FFT spectrum and the advantage and limitation of each algorithm were discussed. For this purpose two experiments were practiced to get the experimental data and these data were used for the modelling of each algorithm and the results were compared and discussed together. In the first experiment, we have practiced impact test for the FRF of the time series modelling which needs two sensor signals in output/input FRF as accelerometer/impact hammer signals. Also, similar experiment for the output/input data of FRF was practiced using the accelerometer and the tool dynamometer sensor for the accelerometer/tool dynamometer FRF respectively.

ARMA (n,m) Model

The relationship between input and output in endmilling operation may be defined as a ARMA (Auto Regressive Moving Average) modelling as follows.

$$\begin{aligned} (1 + a_1 z^{-1} + a_2 z^{-2} + \dots + a_n z^{-n}) y(t) \\ = (1 + b_1 z^{-1} + b_2 z^{-2} + \dots + b_m z^{-m}) e(t) \end{aligned} \quad (1)$$

where a_i ($i = 1, 2, 3, \dots, n$) are auto regressive parameters and b_i ($i = 1, 2, 3, \dots, m$) are moving average parameters. Also $e(t)$ is white noise

AR (n) Model

The AR model can be modelled by using the autoregressive parameters and it can be summarized as follows.

$$(1 + a_1 z^{-1} + a_2 z^{-2} + \dots + a_n z^{-n}) y(t) = e(t) \quad (2)$$

where the detail several algorithm such as burg, least square, yule walker, geometric lattice and instrument variable ar(ivar) algorithms may be modelled like Eq.(2) in such a AR model¹⁾⁴⁾

ARMAX (n,m,l,nk) Model

The ARMAX model is defined as follows and the $A(z)$ in the equation is called autoregressive parameters and the $B(z)$ and $C(z)$ are moving average parameters.

$$\begin{aligned} (1 + a_1 z^{-1} + a_2 z^{-2} + \dots + a_n z^{-n}) y(t) \\ = (b_1 + b_2 z^{-1} + b_3 z^{-2} + \dots + b_m z^{-m}) u(t - nk) \\ + (1 + c_1 z^{-1} + c_2 z^{-2} + \dots + c_l z^{-l}) e(t) \end{aligned} \quad (3)$$

ARX (n,m,nk) Model

The ARMAX model is modelled such as the moving average parameter part $C(z)$ of ARMAX becomes unit by assuming that the past noise has no correlation to current one. So, the following model may be summarized between input and output.

$$\begin{aligned} (1 + a_1 z^{-1} + a_2 z^{-2} + \dots + a_n z^{-n}) y(t) \\ = (b_1 + b_2 z^{-1} + b_3 z^{-2} + \dots + b_m z^{-m}) u(t - nk) + e(t) \end{aligned} \quad (4)$$

Box Jenkins (n,m,l,k,nk) Model

The algorithm may be modelled by assuming that there is different correlation between input and noise input with output. Such model may be represented and also may result in the Box Jenkins model as follows:

$$y(t) = \frac{(b_1 + b_2 z^{-1} + \dots + b_n z^{-n})}{(1 + f_1 z^{-1} + \dots + f_l z^{-l})} u(t - nk) + \frac{(1 + c_1 z^{-1} + \dots + c_m z^{-m})}{(1 + d_1 z^{-1} + \dots + d_k z^{-k})} e(t) \quad (5)$$

Output Error (n,l,nk) Model

By setting the part $C(z)/D(z)$ to be unit in Box Jenkins model by assuming that there is no correlation between past noise and current one, the output error model of time series may be defined as follows.

$$y(t) = \frac{(b_1 + b_2 z^{-1} + \dots + b_n z^{-n})}{(1 + f_1 z^{-1} + \dots + f_l z^{-l})} u(t - nk) + e(t) \quad (6)$$

From the time series models defined above the natural frequency and damping ratio from the denominator of transfer function in each model may be calculated as following.

$$\bar{\omega}_i = \text{abs}(\ln(\lambda_i)) / T_s \quad (7)$$

$$\xi_i = -\cos(\text{angle}(\ln(\lambda_i))) \quad (8)$$

Application to real data

In the first experiment an impulse hammer test was conducted under the condition of no rotation of spindle and the accelerometer signal was measured simultaneously. In the second, An experiment was also practiced to acquire the signal of cutting force and accelerometer in the process of cutting during the endmilling operation.

Frequency Response Function - Impact Hammer Test

Fig.1 (a) and (b) shows the impact hammer signal and accelerometer signal respectively. The right side was magnified for easier analysis.

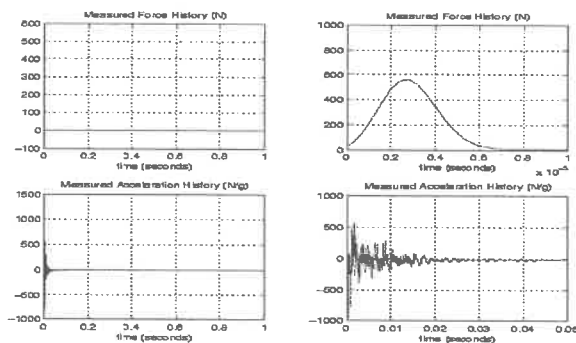
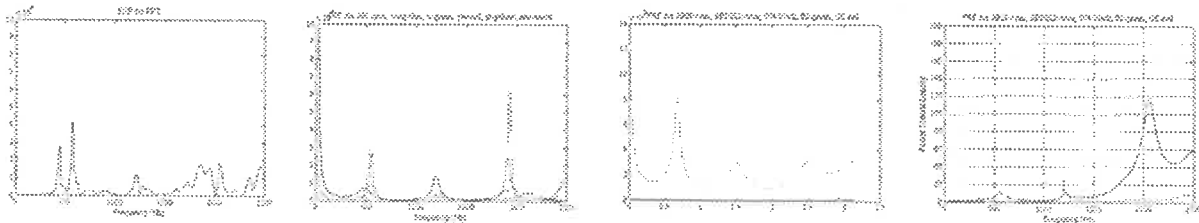


Fig.1 (a) Impulse signal (up - two figure),
(b) Accelerometer signal (down)
for the impact hammer test

The natural frequency and damping ratio were calculated by using Eq.(7)-(8) and they were summarized in Table 1. As a result, the four theoretical natural modes were calculated by Eq.(7)-(8) and the modes are consistent well with its spectrum modes. But the real mode can't be discriminated and just only shows the mode shifting. So it causes a wrong natural mode. If the sampling frequency is selected as 5000 Hz, it

shows a natural mode that is consistent with its real mode a little bit well. The transfer function $H(z) = A_c(z)/F_{imp}(z)$ of the time series models shows that the mode moves to its real one closer. Fig.2(a)~(f) are the results of the power spectrum of each algorithm with real modes about 470 and 600 Hz and they show the smoothing of the modes which are very close to one another. The result of Table 1 is the natural mode calculation with each algorithm. In the table, it was concluded that the calculation modes are consistent well with their real ones and the two nearby modes become into one because of the smoothing characteristics in different parameter selection.



a) FRF for FFT

(b) FRF for AR

(c) FRF for other algorithms
in normalized frequency

(d) FRF for other algorithms
((c) is magnified)

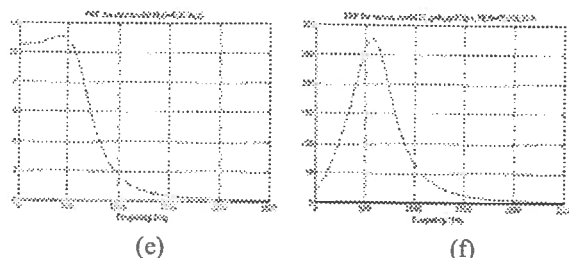


Fig.2 Power Spectrum for each method (a-f) with sampling frequency 5000Hz

The spectrum modes of each line show one of the algorithm and its mode also is consistent well with the mode in Table 1. Because of the different roots in the autoregressive parameters, which consist of the characteristic equation of the model, each algorithm show a little bit different value of its mode. But the shape also gives a good consistency with the real mode and it gives a little bit of mode shifting. Also, by selecting higher sampling frequency, the real frequency for the range 0~1200 Hz may be difficult to identify because of its poor resolution of the lower real range. So it is very important to select the suitable sampling frequency f_q . it was proved that in time series modelling it is enough to sample the data by selecting the sampling frequency f_q below 5000 Hz.

Table 1 Natural mode of structural dynamics

mode	1st	2nd	3rd	4th
ar (ω_i)	5.3617391e+002	1.1971344e+003	1.9248368e+003	
(ξ_i)	3.4378699e-002	5.5848027e-002	1.6478783e-002	
burg	5.5286873e+002	1.2140814e+003	1.9088100e+003	
	3.3994177e-002	6.2746282e-002	2.7552828e-002	
ls	5.2972974e+002	1.1963021e+003	1.9473690e+003	
	9.0083952e-002	6.2663993e-002	1.1828698e-001	
yw	5.5495699e+002	1.2092989e+003	1.8977964e+003	1.7736031e+003
	6.0593751e-002	1.2953545e-001	6.8377535e-002	4.5014260e-001
gl	5.5070053e+002	1.2148282e+003	1.9121992e+003	1.7736031e+003
	3.0100684e-002	5.6566984e-002	2.4355830e-002	4.5014260e-001
ivar	4.9147926e+002	1.1981895e+003	2.1355418e+003	1.8078015e+003
	1.2164688e-001	4.2452551e-002	7.2070567e-002	2.4311292e-001
arx	5.3575944e+002	1.1973181e+003	1.9927695e+003	1.8245005e+003
	4.4241368e-002	7.9502739e-002	9.0351934e-002	5.8455895e-001
armax	1.9792561e+003	5.1048995e+002	1.2435657e+003	1.0247127e+003
	1.1544617e-001	5.7283195e-002	3.8357154e-002	3.4731888e-001
iv4	5.6075439e+002	1.1971848e+003	2.0528913e+003	1.8195641e+003
	6.4704804e-002	2.5695726e-002	4.5152587e-002	1.7867282e-001
bj(c/d)	5.4610182e+002	1.2373347e+003	2.0276169e+003	1.0040063e+003
	6.1545264e-002	5.9670742e-002	8.1668273e-002	6.2406816e-001
bj(b/f)	5.5815957e+002	1.1900335e+003	1.8707502e+003	2.3948874e+003
	1.6086081e-001	3.7050259e-003	8.0266477e-002	6.4016843e-002
oe	4.8063664e+002	1.2004411e+003	1.8833027e+003	2.3464432e+003
	1.4433280e-001	3.1343098e-002	7.6478752e-002	8.3467112e-003
rax	8.3860471e+002	1.5572194e+003	1.8833027e+003	2.3464432e+003
	7.5941489e-002	1.5021026e-001	7.6478752e-002	8.3467112e-003
ramax	8.3482534e+002	1.5773515e+003	1.8833027e+003	2.3464432e+003
	1.5028969e-001	1.6761302e-001	7.6478752e-002	8.3467112e-003

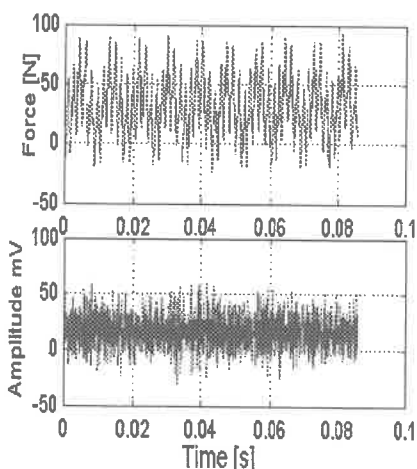
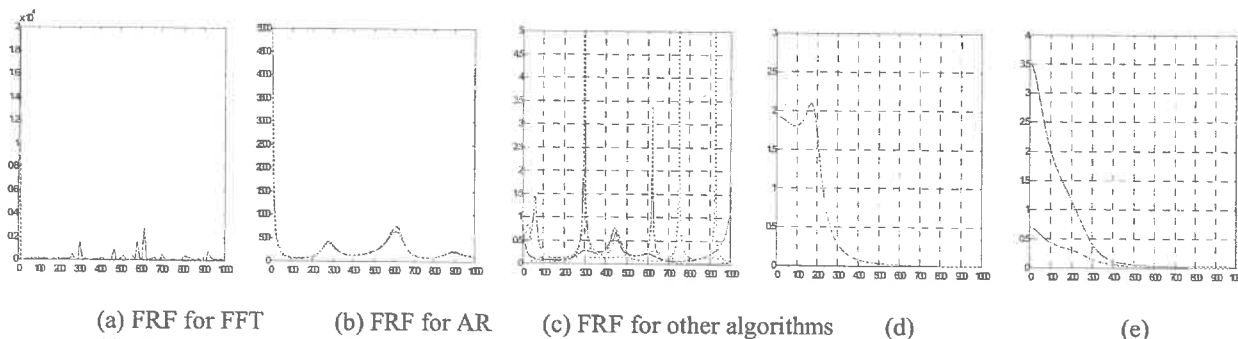


Fig.3 Experimental force and accelerometer signal

Frequency Response Function - Accelerometer/Tool Dynamometer Test



(a) FRF for FFT (b) FRF for AR (c) FRF for other algorithms (d) FRF for rax ($H(z)=B(z)/A(z)$ -up line, $H(z)=1/A(z)$ -down line) (e) FRF for ramax ($H(z)=B(z)/A(z)$ -down, $H(z)=C(z)/A(z)$ -up line)

Fig.4 Power Spectrum for each methods for sampling frequency 2000Hz with Accelerometer / force test.

Fig.3 shows the experimental results that were conducted in the full immersion slot endmilling with the diameter of 15 mm and four flutes endmill, 1500rpm, depth of cut in axial direction of 5mm and feedrate of 750 mm/min

respectively. In any different condition, the two natural frequencies around 470 Hz or 600 Hz may also be got with FFT. Fig.4 shows FRF for the sampling frequency of 4000 Hz. It gives us a good FRF than the case of the high sampling frequency (25000Hz) before. Also, the similar results may be got even if in same condition of impact test with less natural mode deviation. Also the natural mode (frequency, damping ratio) was calculated similarly by using the Eq.(7)-(8) in Table2. But because of the lower order of the model it is very difficult to discriminate the two close modes exactly in the real mode range. It results in the smoothing effect of the weak mode and they are merged into one natural mode. If the sampling frequency is decreased the discrimination of two close modes may be easy with the time series model in this range. The mode discrimination of two close modes was easier for the case of sampling 2000Hz than 4000 Hz by the lowering sampling frequency. This phenomenon may be seen well in Fig.4(c) with the ARX, ARMAX and Box Jenkins algorithm of two real modes about 470Hz and 600 Hz. So it is very important to select the sampling frequency and the time series algorithms appropriately. But using the conventional FFT algorithm it is very difficult to discriminate the close natural modes exactly because of the intrinsic aliasing effect. In general, the ARX and ARMAX are more desirable methods for the time series modelling in endmilling operation than the other algorithms. Also the natural modes were calculated exactly by using Eq.(7)-(8) and summarized in Table 2 for comparing with mode of the spectrum. It shows the exact consistency between two modes, the one by spectrum and the other by theoretical calculation. By lowering the sampling frequency the more precise analysis in the lower frequency range may be acquired in high resolution. Also, the similar mode may be matched for the characteristics of tool passing frequency, tool breakage and chatter, etc. Table 2 is the result of the natural mode that was acquired by setting the sampling frequency as 2000 Hz. In this case the frequency that can be analysed exist in the range of 0 ~ 1000 Hz.

Table 2 Natural mode of Cutting dynamics (sampling frequency = 2000Hz)

mode	1st	2nd	3rd	4th	5th
ar (ω_i) (ξ_i)	2.8046498e+002 1.3798574e-001	8.9801986e+002 1.1272540e-001	6.1909418e+002 9.5336752e-002	5.0351297e+002 2.7934171e-001	
burg	2.8072272e+002 1.3387338e-001	8.9599090e+002 1.1338032e-001	6.1951553e+002 9.7001972e-002	5.0703165e+002 2.8486687e-001	
ls	2.8973510e+002 1.4441778e-001	8.9657807e+002 1.0832206e-001	6.1814192e+002 9.3836287e-002	5.0567556e+002 2.7969961e-001	
yw	2.8299337e+002 1.4748409e-001	8.9568149e+002 1.1493127e-001	6.1970958e+002 1.1056798e-001	5.2411474e+002 2.8942964e-001	
gl	2.8073267e+002 1.3384402e-001	8.9600304e+002 1.1337181e-001	6.1951826e+002 9.6985257e-002	5.0703459e+002 2.8486672e-001	
ivar	8.3207628e+002 3.0407962e-002	6.1219762e+002 1.9393469e-002	2.8316786e+002 2.5940489e-002	3.5512790e+002 8.1214887e-002	
arx	8.9304017e+002 8.3619450e-002	6.3212242e+002 7.0921369e-002	2.7763453e+002 1.6346990e-001	4.6061170e+002 2.5758121e-001	
armax	2.9292011e+002 1.1311945e-001	4.3275066e+002 4.8553468e-002	6.1781229e+002 7.5765545e-002	9.5479606e+002 1.3941328e-001	
iv4	5.8082322e+002 2.6150358e-001	9.0637634e+002 1.6225629e-002	2.5596880e+002 1.7410530e-003	4.5221593e+002 5.0685203e-002	
bj(c/d)	2.8314758e+002 1.1605418e-001	6.2777579e+002 5.8963395e-002	7.9070803e+002 9.9435217e-002	5.4764239e+002 3.6389007e-001	
bj(b/f)	8.8260030e+002 9.8813366e-003	7.0624556e+002 5.3386788e-003	4.3062803e+002 3.7443144e-002	1.2135657e+002 9.9824969e-003	2.8965601e+002 2.5726273e-001
oe	5.6371790e+001 8.8624006e-002	4.2057702e+002 9.1785345e-003	9.1880443e+002 7.1884229e-003	6.6697031e+002 7.7445028e-003	7.2775140e+002 2.2196170e-001
rax	6.5316988e+002 1.0976963e-001	3.5047675e+002 2.1283801e-001	9.1880443e+002 7.1884229e-003	6.6697031e+002 7.7445028e-003	7.2775140e+002 2.2196170e-001
ramax	4.7144256e+002 2.4839496e-001	7.0537816e+002 1.3504902e-001	9.1880443e+002 7.1884229e-003	6.6697031e+002 7.7445028e-003	7.2775140e+002 2.2196170e-001

Results

Among several algorithms, ARX and ARMAX models were proved to be suitable for the natural mode analysis in endmilling and these may be used in the more reliability for getting the natural mode with less error. These algorithms give us good calculated results of real natural mode ranging from 470 Hz to 600 Hz with less bias but these several time series methods also give us good natural modes in lower sampling frequency about 2000 Hz especially in ARX and ARMAX model. But it is impossible to get the natural mode in low sampling frequency with FFT method. In natural mode estimation the FRF $A_c(z)/F_{imp}(z)$ of impact hammer test and $A_c(z)/F_{dyn}(z)$ of dynamometer test give us the same natural mode. So the latter method can be used easily than former one with lower sampling frequency for natural mode estimation and other malfunction mode such as chatter and tool passing frequency.

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Defining the measurand in radius of curvature measurements

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The radius of curvature of an optical element is one of the dominant parameters that determines optical power. Radius errors can be partially compensated by element respacing, but this is time-consuming and costly and can lead to unwanted wavefront aberrations. For spherical components, the radius is usually defined as the radius of the best-fit sphere over the clear aperture of the lens in a least-squares sense. Many measurement methods are available, including comparison with 'known radius' test plates, direct image formation/knife-edge test, astigmatism measurement, spherometers, autocollimator with pentaprism, and shearing interferometers^{1,2}, but the optical radius bench generally gives the lowest uncertainty. The radius is measured on an optical bench by using a figure measuring interferometer to identify the location of the test artifact at two critical positions, confocal and cat's eye. The reflected wavefront matches the wavefront exiting the interferometer at these two locations, resulting in a nulled cavity. A displacement gauge measures the displacement of the part between the two positions, and the distance is nominally the radius of the best-fit sphere to the surface. To carry out a traceable radius measurement, however, all biases must be corrected and all uncertainty sources identified and combined to yield an overall combined standard uncertainty - a formidable task for this deceptively simple measurement.

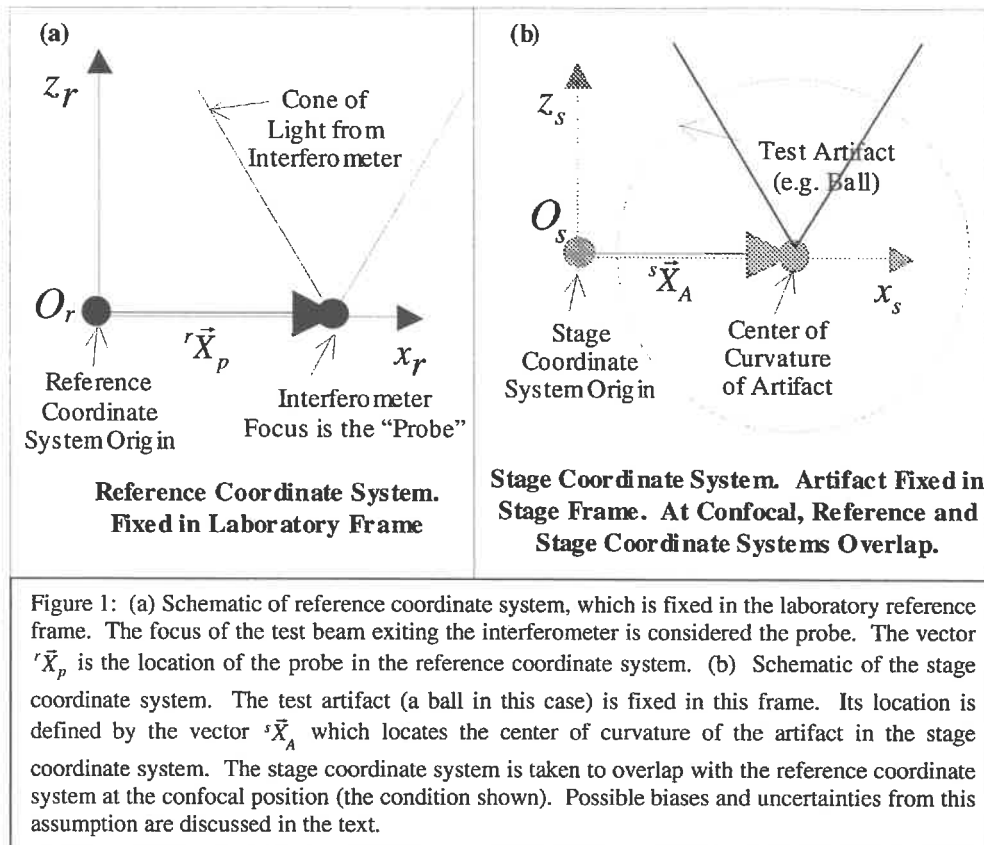
There are many sources of uncertainty and measurement bias, as discussed in the literature³⁻⁵, such as wavefront aberrations, identification of null at the two locations, figure error corrections, displacement gauge reading, and stage error motions. Error motions are particularly challenging because they are interrelated and convolved with the measurement. Recent work has focused on identifying uncertainty contributions, removing measurement biases, and estimating the uncertainty through a root-sum-of-the-squares (RSS) combination^{4,5}. However, a mathematical formalism has yet to be presented which defines the radius as a function of all uncertainty sources. The radius is not a simple linear function, and a mathematical model is needed to ensure that all uncertainty sources are properly combined.

Here we report on a method of mathematically defining the measurand, R , for optical bench radius measurements that yields a single function that intrinsically corrects for error motions and also allows other uncertainty sources to be represented. The method is based on a homogeneous transformation matrix (HTM) formalism. The formalism is well known in the precision engineering community and is commonly used to analyze error motions in precision machine tools and stages⁶. We use a vector equation to define the radius and an HTM to determine the location of the test artifact after translation from confocal to cat's eye. To use our results, the test engineer must first characterize the error motions of the radius bench and assess uncertainties, then calculate a corrected estimate of the radius and its uncertainty based on the formalism described here.

In order to correct for stage error motions, the final location of the test artifact after translation must be determined through a coordinate transformation and this requires knowledge of the rigid-body motion of the stage. The uncertainty goal and the extent of the error motions determine the level of sophistication with which the coordinate mapping must be done. One can consider three general categories:

- (i) The error motions are zero on average. Surprisingly, even in this limit, the simple reading of the gauge at cat's eye compared to confocal is not a good estimate of the radius. Error motions, either positive or negative, always lead to a gauge reading that is less than the actual radius, and this introduces a bias in the measurement, even if the error motions are zero on average⁷. Our formalism intrinsically corrects for this bias.
- (ii) The error motions increase linearly with displacement.
- (iii) The error motions are nonlinear functions of the displacement. This is always the case at some level. This limit must be considered in very demanding uncertainty applications or where the instrument is poorly designed and/or fabricated.

Only the rigid body motion that occurs *during* the displacement is important; consequently, the absolute position of the stage is not critical for categories (i) and (ii), but is for category (iii). The results presented here apply to categories (i) and (ii). Category (iii) requires a more sophisticated treatment.



which is referred to as the reference coordinate system. We define the z-axis of the reference frame as the axis of the displacement gauge. Because we are considering categories (i) and (ii) only, the absolute z-coordinate of the probe or artifact is irrelevant, consequently we set each to zero. The two coordinate systems are shown in Figure 1. The probe vector, $r\vec{X}_p$, is defined in the gauge calibration process, which defines the reference z-axis orientation and location. The location of the artifact on the stage is defined at the starting position for the measurement – the confocal position. Because we restrict our analysis to categories (i) and (ii), we can define the stage and reference coordinate systems to exactly overlap at confocal. Consequently the vectors $r\vec{X}_p$ and $s\vec{X}_A$ are equal at this point, but only up to the ability to null the interferometer at confocal. This gives us $s\vec{X}_A = r\vec{X}_p + (dx^{cf}, dy^{cf}, dz^{cf})$, where the last vector represents nulling errors.

Figure 2 illustrates a radius measurement – translation from the confocal to the cat's eye position. Current practice is to equate the radius with the gauge reading (assumed zeroed at the starting confocal position). This corresponds to the projection of the center of the artifact onto the z-axis at cat's eye. As illustrated in the figure with x-axis straightness error in the motion, the gauge reading at cat's eye (d_{ce}) is less than the radius, consequently the measurement is biased. Rather than define the radius in terms of the projection, it can be exactly defined with the appropriate vectors, as shown in the figure. If the location of the artifact at cat's eye is known in the reference frame, the vector defining the radius is given by $\vec{R} = r\vec{X}_p - r\vec{X}_A^{ce}$, and the magnitude of $\vec{R}$ is given by

$$R^2 = |\vec{R}|^2 = |r\vec{X}_p - r\vec{X}_A^{ce}|^2.$$

To determine the location of the artifact at cat's eye, the rigid body motion of the stage must be known. With this knowledge we can map the coordinates of the artifact back to the reference coordinate system by using an HTM⁷. An HTM is a 4 x 4 matrix that captures the rotation and/or translation of one coordinate system relative to another and allows a vector in one system to be mapped to the other. In general, matrices that represent rotation about orthogonal axes do not commute, but they do in the limit of small rotations⁷. An assumption of small rotations is reasonable for a well-designed radius bench, and in this limit, a general HTM for a nominal displacement of d along the z-axis is given by⁷

The method requires the definition of two coordinate systems and two critical vectors. The two systems are a system fixed to the stage and a one fixed to the focus of light exiting the interferometer that completes the metrology loop. The test artifact (e.g. a ball) is rigidly attached to the stage and there is a unique vector that defines its location, $s\vec{X}_A$, taken to be the center of curvature. The location of the beam focus is the other critical vector, $r\vec{X}_p$.

The beam focus, or probe, is fixed in the laboratory frame,

(1)

After substituting into our vector equation defining R , we have

(2)

With substitution of Equation 1 for the HTM and following the multiplication, we can write out the components of $\vec{R}$. We need to calculate the magnitude of the vector $\vec{R}$, which involves squaring and summing the three coordinates. Grouping similar terms we end up with

(3)

This equation is used to estimate the radius from a measurement by evaluating the expression with the expectation values of each term. Our measurand, R , is equal to the square root of this expectation value, namely $R = \sqrt{\langle R^2 \rangle}$. We have assumed that all parameters are uncorrelated.

At first Equation 3 appears daunting, but most of these terms are zero with a well-designed radius bench. For the case where all error motions are zero on average and the gauge axis is collinear with the probe (with some uncertainty), the equation reduces to

(4)

Note that the expectation values of the squared terms are not zero – these expectations are equal to the variance of the parameter⁸. The higher order terms may be significant in some cases. The presence of the variances in addition

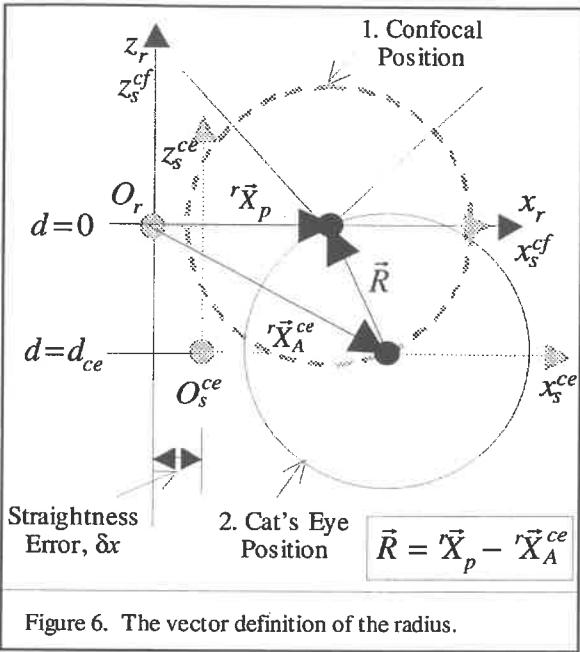


Figure 6. The vector definition of the radius.

to the d_{ce}^2 term reflects the biases that are intrinsically present because all misalignments and errors always lead to a value of d_{ce} that is too small.

An equation for the measurand makes it possible to use a Taylor series expansion to directly calculate an estimate of the combined standard uncertainty. Rather than calculating the series expansion of the equation for R , it is easier to work with the series expansion of R^2 first. We can estimate a combined uncertainty for R^2 and then apply error propagation to estimate the uncertainty for R . Letting $x = \langle R^2 \rangle$ then $R = \sqrt{x}$. Applying error propagation we then have^{9, 10}

$$\mu_c = \mu_R = \frac{d(R)}{d(x)} \delta(x) = \frac{1}{2\sqrt{x}} \delta(x) = \frac{\delta(x)}{2R} = \frac{\mu_{R^2}}{2R} \quad (5)$$

Calculating the uncertainty for R^2 can also be daunting. Even though it is a simple sum of terms, the occurrence of products of variables complicates matters, particularly if the expected value of many terms is zero. The simple first order Taylor series expansion may not be enough. Higher-order partial derivative terms may have to be considered. Fortunately, the terms requiring multiple partial derivatives will likely be small and can be ignored. When the error motions and misalignments are zero on average and only the first derivative terms are significant, the combined uncertainty for R^2 becomes

$$\mu_{R^2}^2 = \left(\frac{\partial(R^2)}{\partial d_{ce}} \right)_{\langle d_{ce} \rangle}^2 \mu_{d_{ce}}^2 + \left(\frac{\partial(R^2)}{\partial (dz^{cf})} \right)_{\langle dz^{cf} \rangle}^2 \mu_{dz^{cf}}^2 \quad (6)$$

Example: The complexity of evaluating the expectation value of Equation 3 and estimating the uncertainty will be system dependent. We have applied the new analysis method to a radius measurement of a micro-sphere on a micro-interferometer¹¹. For this instrument, the error motions, probe offsets, and confocal misalignments were all zero on average and uncertainties in these quantities were estimated from details of the alignment, calibration, and measurement procedures. Because error motions and misalignments are zero on average we can use Equation 4 to estimate R^2 and therefore R . From consideration of our uncertainties, it is clear that the higher-order terms in Equation 4 are small and can be ignored. The uncertainty for R^2 is estimated through a Taylor series expansion of Equation 3, and for our case Equation 6 can be used. The best estimate for the radius is 0.408 mm $\pm$ 0.007 mm, which represents a 1.7% combined standard uncertainty. The uncertainty is dominated by the repeatability of the measurement and the calibration of the displacement gauge.

Acknowledgements: We would like to thank Chris Evans at Zygo Corporation for his feedback and comments on the work, NIST for partial funding, and S. Smith, R. Hocken, and M. Davies at UNC Charlotte for helpful discussion about homogeneous transform matrices and coordinate mapping.

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ALTERNATIVE ARTIFACTS FOR EVALUATING SCANNING CMM PERFORMANCE

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1. Introduction

Coordinate measuring machines (CMMs) with continuous-contact scanning capabilities are being sold in greater and greater numbers. The following claim is representative of those touted by manufacturers: "No matter what types of feature controls you are confronted with, high data density scanning is your avenue to finding the true value". A necessary step in comparing scanning CMMs to each other, or to discrete point CMMs, is to use a standardized test method. The current ISO 10360-4 standard [1] describes circular scans on a calibrated sphere. As this sphere is of good quality, the certification test is more an exercise of the CMM motion controller than a test of the behavior of the measuring system under the excitation of measuring a real part surface. A different test has been proposed, that involves scanning over a gage pin with the same diameter as the probe stylus. This test results in a 90 degree turn in the probe path which, although more challenging than a sphere measurement, may not be representative of actual scanning. Past research [2,3,4] at UNC Charlotte has found that by scanning lobed cylindrical artifacts we may excite the resonant frequency of the probing system, resulting in severe performance degradation as the feature is scanned. Our current work examines the role of linear artifacts with a periodic (sinusoidal) waveform superimposed on a flat surface. By measuring these artifacts along different machine axes, and at different speeds, we can excite different frequencies in the probing system and in the machine structure. Our goal is to develop a means of distinguishing between scanning CMMs based on their ability to measure different spatial frequencies and amplitudes at different speeds. This paper describes our work in developing and measuring artifacts that will better capture how a scanning CMM will perform when measuring actual parts.

2. Current Work

To date, the linear sinusoidal artifact (Fig. 1) has been measured to study the performance of the CMM by varying different parameters, such as:

- Scanning speed
- Data density
- Direction of the scan
- Position and orientation of the artifact within the machine volume
- Stylus diameter and probe type

3. Scanning the Sine Artifact

The part coordinate system is constructed by defining the datum planes as follows:

- The Primary Datum (z) is constructed from the flats at each end of wavy section.
- The Secondary Datum (y) is constructed along midplane of the artifact.
- The Tertiary Datum (x) is established at end of the "cutout."

The sine artifact is mounted on the CMM in three different positions. In the first position, the part coordinate system is parallel to the machine coordinate system; in the second position, the part (with its coordinate system) is rotated 45° about machine z-axis; in the third position, the part is rotated 45° about Machine z-axis and inclined at an angle of about 45° from the horizontal (machine xy-plane). In every case, the probe stylus was oriented "straight down" in the minus-z direction.

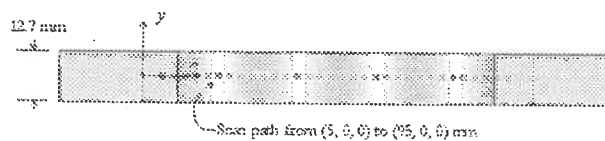
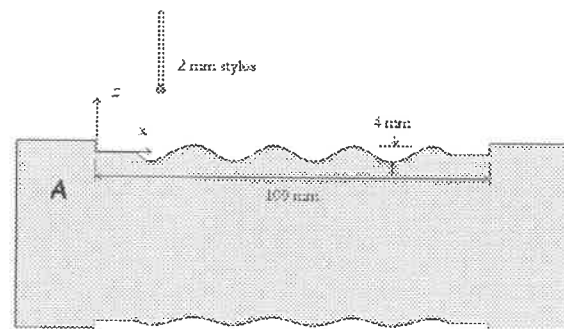


Fig. 1. Sinusoidal Artifact

The artifact is scanned along the path shown in Fig. 1. This scan path is from (5, 0, 0) to (95, 0, 0) mm in the part coordinate system. The wavy (sinusoidal) portion of this path is from $x = 10$ mm to $x = 90$ mm.

4. Data Analysis

In order to remove errors in the measurement of the Primary Datum surface, we first refit the $z=0$ line to the initial and final segments of the scan. Typical values for the slope of this line are between 0 and $8 \mu\text{m}/\text{m}$, and are thus virtually negligible (a slope of $8 \mu\text{m}/\text{m}$ corresponds to an angle of 0.00046°). The measurement data are rotated through this angle. A least square sine wave is then fitted to the wavy portion of the data. MATLAB's optimization toolbox is used to fit these data to the following equation:

$$LS \text{ Sine Wave} = A - B \sin\left(2\pi C \frac{x - D}{E - D}\right)$$

where

A	=	Baseline (Vertical Offset)
B	=	Amplitude
C	=	No. of waves
D	=	x Start Point
E	=	x End Point

The least squares (LS) fit of the sine wave may not have a zero mean line; therefore parameter A controls the vertical position of the LS sine wave. Parameter B controls the amplitude of the LS sine wave, which is nominally 2mm. Parameter C is the number of waves in the artifact; ideally 4 waves. Parameters D and E control the horizontal position of the LS sine wave.

Deviations from the least square sine wave were determined by using:

$$\text{Deviations} = \text{Rotated Data} - \text{LS Sine Wave}$$

Form error of the artifact was determined by:

$$\text{Form Error} = \text{Maximum Deviation} - \text{Minimum Deviation}$$

5. Results

The artifact was scanned using six different scanning speeds: 1, 2, 5, 8, 10 and 20 mm/sec. Data density was set at 10 points/mm. Results in Table 1 and Table 2 corresponds to scans taken parallel to the machine x-axis. It could be observed from Table 1, that the form error is in the range of 14.7 to $15.7 \mu\text{m}$. From this data, we see that change in scanning speed did not result in a significant change in the form error.

Table 1. Effect of different scanning speeds on the form error (FE)

SCANNING SPEED	DATA POINTS	LEAST SQUARE SINE WAVE PARAMETERS					DEVIATION FROM LS SINE WAVE		
		A	B	C	D	E	MAX	MIN	FE
mm/sec	points	μm	mm	waves	mm	mm	μm	μm	μm
1	971	-0.9	1.9982	3.9865	10.0013	89.7293	8.4	-6.7	15.2
2	974	-0.7	1.9982	3.9875	10.0025	89.7493	8.3	-6.5	14.7
5	974	-1.0	1.9982	3.9991	10.0026	89.9822	8.6	-6.9	15.5
8	974	-1.1	1.9983	3.9871	10.0026	89.7427	8.8	-7.0	15.7
10	964	-1.7	1.9983	3.9963	10.0019	89.9266	9.0	-6.7	15.6
20	918	-2.3	1.9983	3.9885	10.0018	89.7697	8.0	-6.9	14.9
Nominal Values		0	2	4	10	90			

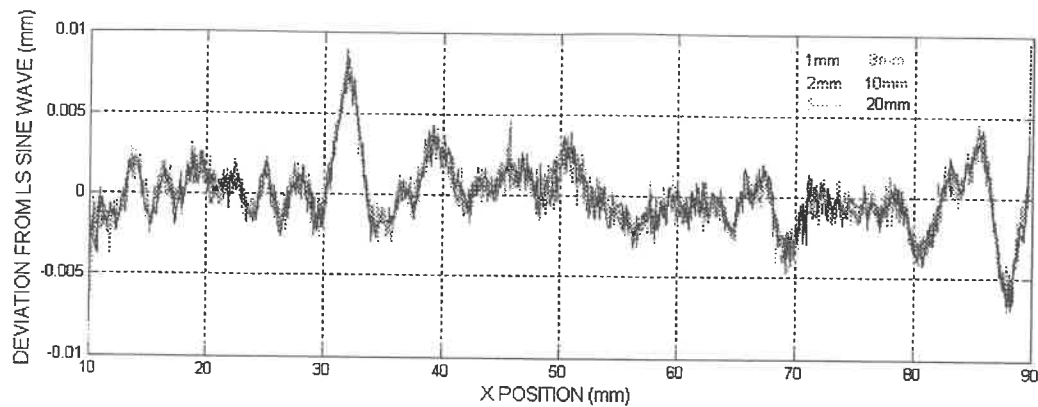


Fig. 2. Deviations from LS sine wave for different scanning speeds

To study the repeatability of the parameter-fitting methods, five scans were repeated with a scanning speed of 2 mm/sec. From Table 2, the average form error (with respect to a fitted sine wave) is 15.4 μm with a standard deviation of 0.6 μm . Ten repeated scans of the artifact on a second CMM at UNCC resulted in an average form error of 14.3 μm with a standard deviation of 0.6 μm .

Table 2. Results from five repeated scans

SCANNING SPEED	SCAN NO.	DEVIATION FROM LS SINE WAVE		
		MAX	MIN	DIFF
2 mm/sec	1	9.0	-6.1	15.1
	2	9.4	-6.9	16.4
	3	8.8	-6.9	15.7
	4	9.1	-6.1	15.2
	5	8.3	-6.5	14.8
		Average		15.4
		St. Dev.		0.6

A second comparison method, more directly related to CMM performance, involves starting with a reference shape for the wavy surface, and then comparing different scans to this reference scan. This allows us to see the effect of different scanning speeds and probe setups without fitting a least squares sine wave each time. Fig. 3 shows a comparison of scans with 2, 5, 8 and 10 mm/sec scanning speeds (scan direction: X). Fig. 4 shows the comparison of a 20 mm/sec scan in X direction with the reference scan. In each of these figures, the reference scan is an average of ten scans obtained on UNCC's Leitz PMM with 1 mm/sec scanning speed in X-direction.

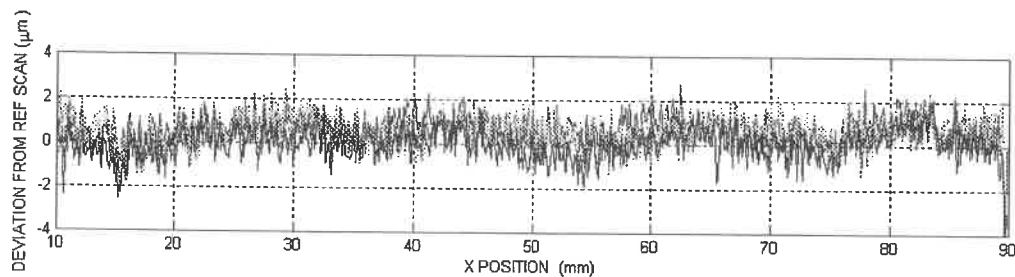


Fig. 3. Scans with 2, 5, 8, 10 mm/sec scanning speed compared to the reference scan

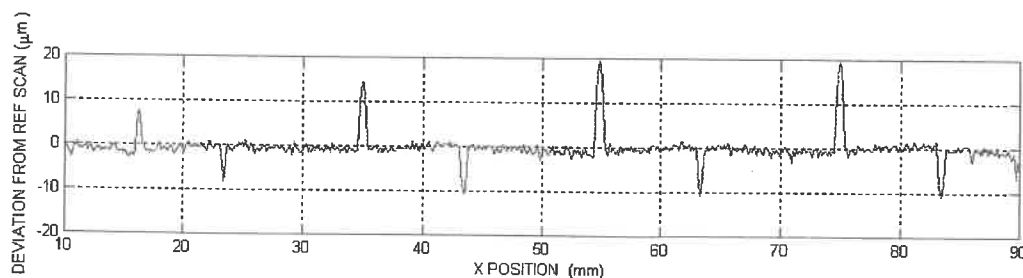


Fig. 4. Comparison of a 20 mm/sec scan in X direction with the reference scan

Fig. 5 shows the comparison of scans in XY and XYZ direction, both with a scanning speed of 1 mm/sec. Fig. 6 shows the deviations from least square sine wave for 2 mm/sec scan in XYZ direction. By comparing Fig. 2 and Fig. 6, we observe that when scanning in XYZ direction, a small increase in the scanning speed may result in a significant change in the deviations.

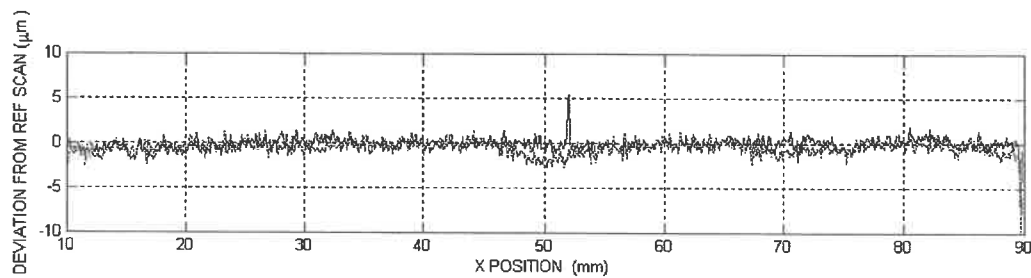


Fig. 5. Scans in XY and XYZ direction compared to the reference scan in X direction (scanning speed: 1 mm/sec)

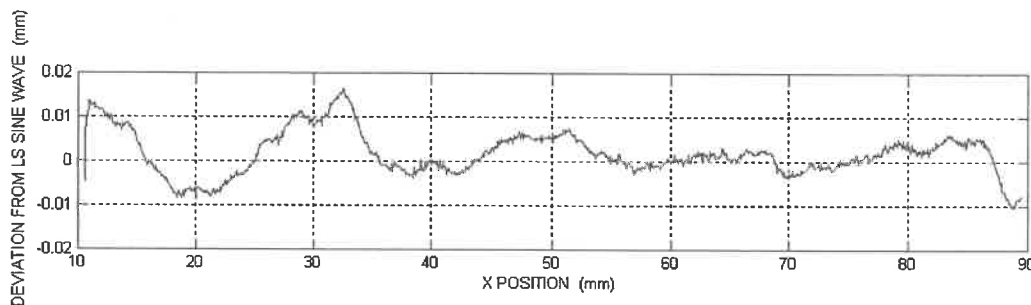


Fig. 6. Deviations from LS sine wave for 2 mm/sec scan in XYZ direction

6. Future Work

The data analysis is currently done without using any kind of external filters. Using a suitable high frequency filter may reduce the noise in the data. This would enable us to compare the data taken on different CMMs. In order for this comparison, we would consider one of the scans as a reference scan. All the other scans will be compared to the reference scan. Another sinusoidal artifact is being considered instead of the linear sinusoidal artifact (Fig. 7). This artifact would have a better surface finish.

Acknowledgements

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Fig. 7. Another sinusoidal artifact

Study on Support Method in Sori Measurement of 300mm Silicon Wafer

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1 Introduction

The “sori” of a semiconductor wafer is defined as the difference between the maximum and minimum distances to the front surface of a free, unclamped wafer from the least squares reference plane of the front surface. The sori measurement of a large diameter silicon wafer, for example 300mm wafer, at nanometer order is required in order to achieve the maximum yield of semiconductor device processing. Also, sori measurement is used to monitor residual stress in wafers caused by thermal and mechanical effects during device processing. It is well-known that precise sori measurement for silicon wafers of 300mm in diameter and 0.775mm in thickness is difficult due to gravity-induced deflection and other disturbances. Up to now, the American Society for Testing and Materials (ASTM) has proposed a sori measuring method [1], called one-point-support method in this study because the wafer is supported by a small-area chuck at the wafer center during the measurement. The shape of the front surface including gravity-induced deflection is measured with the front surface upward, and then it is measured again after turning the front surface downward. Sori is thus obtained from the two measurements using the principle of the gravity cancel method by wafer inversion [2]. However, the use of only one support at the wafer center makes the measuring process unstable, resulting in decreased repeatability. Especially, when the wafer is vacuum chucked, the shape in the absorption area cannot be measured and the sori itself may vary depending on degree of vacuum. On the other hand, a method for measuring the bow of a silicon wafer under 100mm diameter has been specified by the Japanese Industrial Standard (JIS) [3]. With this method, the bow, which is how concave or convex the deformation of the median surface of the wafer is at its center point, is obtained by taking the difference between heights of the wafer at its center point measured with the wafer front surface upward and downward, when the wafer is horizontally supported by three balls. Because the bow is obtained only from the measured data at the wafer center, the measured value is different from sori. Besides, the wafer shape over the whole surface cannot be obtained. Finally, there is a method [4] with which the wafer is set vertically in order to decrease the effects of gravity-induced deflection. The wafer is supported at three positions on the periphery and rotated during the measurement. It is obvious that the force working on each support changes with the rotation of both wafer and support so that the wafer position may move slightly during measurement. In addition, buckling and distortion of wafer are thought to be caused by clamp force, making accurate sori measurement impossible.

In this study, a new sori measurement method called three-point-support method was proposed. With this method, the gravity cancel method mentioned above is applied to a 300mm silicon wafer supported horizontally by three steel balls on the vertexes of a regular triangle at the wafer edge. A comparison between the one-point-support and three-point-support methods was carried out experimentally, considering the repeatability influenced by wafer loading operation, the change in the degree of vacuum, and disturbance vibration. Finally, the influence of the crystal orientation on the sori measurement was investigated.

2 Measurement principle and experimental equipment

The 300mm wafers used in this research were single crystal silicon wafers, whose front and back surfaces were polished to mirror-surface finish. The thickness deviation of the wafers was about 1 μ m, which is relatively small compared to the sori itself.

2.1 Principle of sori measurement

In this study, sori was measured with the gravity cancel method by wafer inversion [2]. The principle of this method can be explained using the one-point-support model shown in Fig.1. The principle was also similarly applied to a three-point-support model. Here, the condition in which the front surface is faced upward is called the normal direction, while that in which the front surface is faced downward is called inverted direction. After the shape of the wafer front surface $y_f(x,y)$ including gravity-induced deflection $g_f(x,y)$ and true shape $s_f(x,y)$ in normal direction, as expressed by equation (1), is measured, the wafer is turned in the inverted direction and the shape of the back

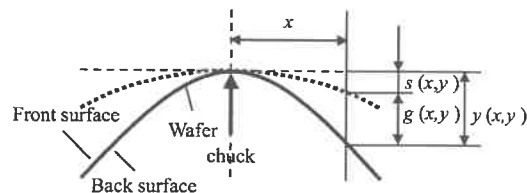


Fig.1 Principle of sori measurement

The artifact is scanned along the path shown in Fig. 1. This scan path is from (5, 0, 0) to (95, 0, 0) mm in the part coordinate system. The wavy (sinusoidal) portion of this path is from $x = 10$ mm to $x = 90$ mm.

4. Data Analysis

In order to remove errors in the measurement of the Primary Datum surface, we first refit the $z=0$ line to the initial and final segments of the scan. Typical values for the slope of this line are between 0 and 8 $\mu\text{m}/\text{m}$, and are thus virtually negligible (a slope of 8 $\mu\text{m}/\text{m}$ corresponds to an angle of 0.00046°). The measurement data are rotated through this angle. A least square sine wave is then fitted to the wavy portion of the data. MATLAB's optimization toolbox is used to fit these data to the following equation:

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where

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B	=	Amplitude
C	=	No. of waves
D	=	x Start Point
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The least squares (LS) fit of the sine wave may not have a zero mean line; therefore parameter A controls the vertical position of the LS sine wave. Parameter B controls the amplitude of the LS sine wave, which is nominally 2mm. Parameter C is the number of waves in the artifact; ideally 4 waves. Parameters D and E control the horizontal position of the LS sine wave.

Deviations from the least square sine wave were determined by using:

$$\text{Deviations} = \text{Rotated Data} - \text{LS Sine Wave}$$

Form error of the artifact was determined by:

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5. Results

The artifact was scanned using six different scanning speeds: 1, 2, 5, 8, 10 and 20 mm/sec. Data density was set at 10 points/mm. Results in Table 1 and Table 2 corresponds to scans taken parallel to the machine x-axis. It could be observed from Table 1, that the form error is in the range of 14.7 to 15.7 μm . From this data, we see that change in scanning speed did not result in a significant change in the form error.

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10	964	-1.7	1.9983	3.9963	10.0019	89.9266	9.0	-6.7	15.6
20	918	-2.3	1.9983	3.9885	10.0018	89.7697	8.0	-6.9	14.9
Nominal Values		0	2	4	10	90			

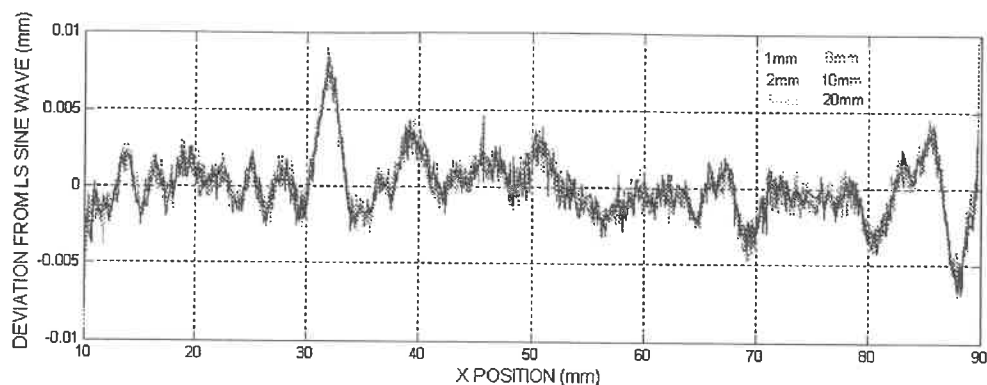


Fig. 2. Deviations from LS sine wave for different scanning speeds

To study the repeatability of the parameter-fitting methods, five scans were repeated with a scanning speed of 2 mm/sec. From Table 2, the average form error (with respect to a fitted sine wave) is 15.4 μm with a standard deviation of 0.6 μm . Ten repeated scans of the artifact on a second CMM at UNCC resulted in an average form error of 14.3 μm with a standard deviation of 0.6 μm .

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	4	9.1	-6.1	15.2
	5	8.3	-6.5	14.8
Average				15.4
St. Dev.				0.6

A second comparison method, more directly related to CMM performance, involves starting with a reference shape for the wavy surface, and then comparing different scans to this reference scan. This allows us to see the effect of different scanning speeds and probe setups without fitting a least squares sine wave each time. Fig. 3 shows a comparison of scans with 2, 5, 8 and 10 mm/sec scanning speeds (scan direction: X). Fig. 4 shows the comparison of a 20 mm/sec scan in X direction with the reference scan. In each of these figures, the reference scan is an average of ten scans obtained on UNCC's Leitz PMM with 1 mm/sec scanning speed in X-direction.

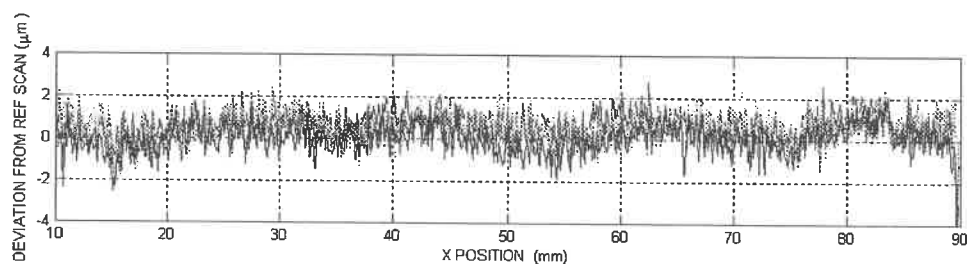


Fig. 3. Scans with 2, 5, 8, 10 mm/sec scanning speed compared to the reference scan

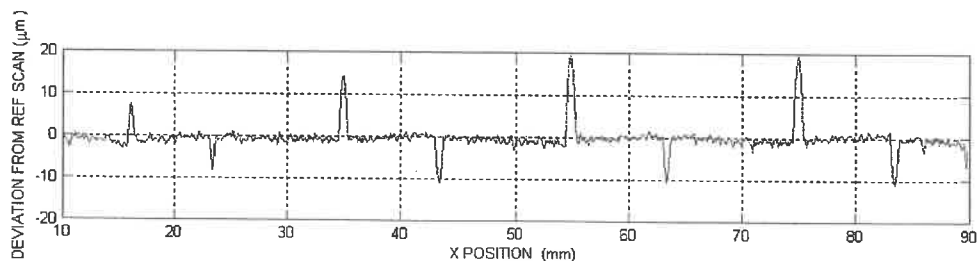


Fig. 4. Comparison of a 20 mm/sec scan in X direction with the reference scan

The sori measured value changed greatly from 15 μm to 25 μm after the support positions were moved (see Fig.8). This difference proves that the positioning accuracy of the supports plays an important role in the three-point-support method.

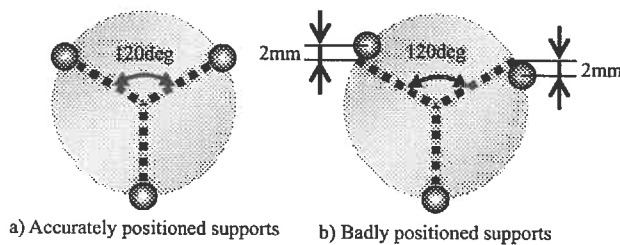


Fig.7 Arrangement of supports in experiments investigating effects of positioning accuracy

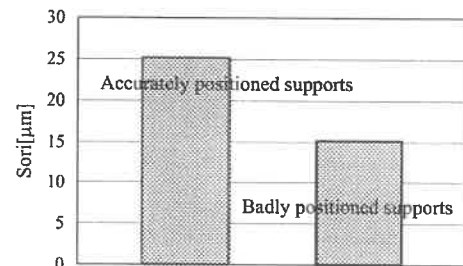


Fig.8 Results of effects of support positioning accuracy on sori measurement

5 Effect of crystal orientation on sori measurement

As mentioned in section 2.1, in order to improve the measuring efficiency, subtracting the gravity-induced deflection of a representative wafer from the measured results of a sample wafer in normal direction is sometimes used in sori measurement. However, even if the sample wafer has the same diameter and thickness as the representative wafer, a large error will occur if the crystal orientations of these two wafers are different. This is because elastic stiffness constant depends on the crystal orientation. Here, the effects of crystal orientation on the gravity-induced deflection are discussed. The wafer was supported by a chuck 30mm in diameter on the center, then the gravity-induced deflection distribution was obtained with the gravity cancel method using equation (6). The contour map of the gravity-induced deflection in showed distorted curves especially towards the circumference, and they are symmetric to the axes with 45 degree interval. The maximum deflections of the wafer on the cross section every 45 degrees are shown in Fig.9. It was found that the maximum deflection changes about 7 μm every 45 degrees, which means that the deflection of silicon wafer changes with orientation. This result shows that sori measurement result will bear a error about several μm , if the orientation of the sample wafer differs from that of the representative wafer, not only in one-point-support but also in three-point-support when sori is measured using the representative wafer.

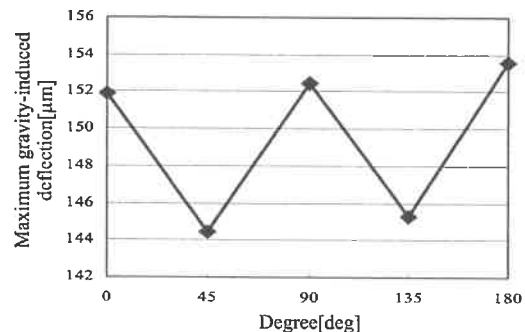


Fig.9 Change in maximum gravity-induced deflection with crystal orientation

6 Conclusions

In this research, a three-point-support method for the sori measurement of a 300mm silicon wafer was proposed. From the comparison of the repeatability and anti-disturbance ability between the proposed method and the conventional one-point-support method, it is found the three-point-support method is superior to the conventional one in repeatability and anti-disturbance ability. Moreover, precise positioning of the supports is crucial in sori measurement with the three-point-support method. Finally, a measuring error about several μm will appear if the crystal orientation of the sample wafer is different from that of the representative wafer when sori is measured using the representative wafer.

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OPTICAL POLYGON CALIBRATION USING ONLY ONE AUTOCOLLIMATOR

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ABSTRACT

In this paper we present a new measuring method to obtain the angular variations of the faces of an optical polygon using only one autocollimator. The method requires additionally an external beam splitter to divide the light from autocollimator and a flat mirror for second autocollimator emulation. The method can be used for polygons of any number of faces. In this case we present calibration results of a twelve faces polygon.

1. INTRODUCTION

Polygon mirrors are used as angle standards for angular calibrations, for example to determine the angular displacement accuracy of machine tool rotary tables. There are two well-known methods for polygon mirror calibration stated in the industrial standards [1] [2]. One of them uses two optical polygons with the same number of faces (or at least an integer multiple) and one autocollimator. This method compares errors between both polygons; from this comparison it is possible to separately determine errors of each. The other method uses only one polygon but two autocollimators. In this case, the angle between every adjacent pair of faces of the polygon is compared to the angle formed by the two autocollimators.

The main problem we found for the calibration of the polygon is concerned to our restricted resources; we just have one polygon and one autocollimator. So we need to develop a method adequate to these resources.

In "ASPE 1996 Annual Meeting" [3] we proposed a method that uses a He-Ne LASER beam, a beam splitter, a mirror, a lens, a knife-edge and a photodetector. Besides, it requires a previous calibration of photodetector response correlation between beam intensity and angular displacement for two beam trajectories and was not precise enough for our calibration.

In this paper we present a different method to overcome the above mentioned problems. The calibration procedure uses only one autocollimator for calibrating only one polygon at a time, in this case the optical setup is similar as that in ref. [3]. It can be used to measure any kind of optical polygon with flatness of $\lambda/10$ or better on the reflective faces.

2. METHOD

2.1 Experimental set up

The optical polygon is placed on a flat plate with a spindle or on a turntable, in order to let free turn of polygon. An autocollimator is aligned over the face 0°. After that, a beam splitter is

placed between both devices obtaining “A” and “B” beams. One of them, *A*, goes directly towards the 0 degree face of polygon, while the other, the beam *B*, is directed towards an adjacent face aided through a flat mirror as showing in Fig. 1a).

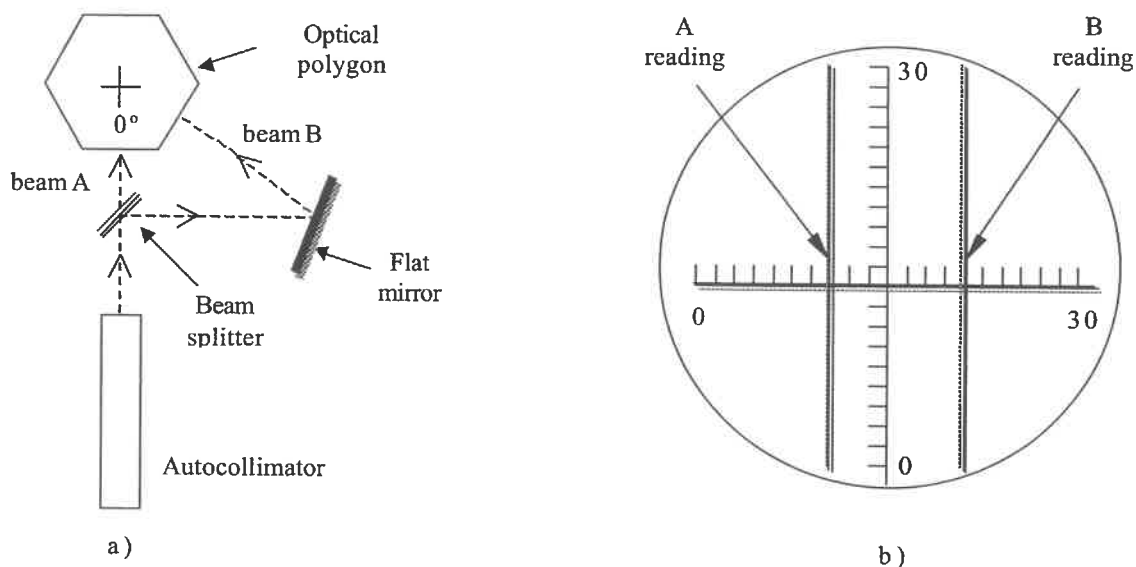


Figure 1-. a) Optical device showing divided beams “A” and “B”. b) View inside of autocollimator, position of “A” and “B” readings.

Now, as showing in Fig. 1b), inside of autocollimator we have two readings, one from each beam. It is convenient to have low readings for beam *A* and high readings for beam *B* to effect of avoid confusion with both readings. In the case of electronic autocollimators, it is possible to block the beams in alternative way in order to read each separately.

2.2 Experimental procedure

Once the devices have been aligned as described above, both readings are taken, *i.e.*, *A* reading from 0° face and *B* reading from adjacent face. For this particular case, we have a twelve faces optical polygon, so *B* reading comes from 30° face.

After that, the optical polygon is revolved until beam *A* aims to 30° face and beam *B* towards 60°. *A* and *B* readings are taken again. This procedure is repeated until all other pairs of adjacent faces are measured.

2.3 Formulas for computing the angular variation of the polygon faces

Having the angles $\theta_1, \theta_2 \dots \theta_n$ for each face of the polygon, all of them measured from 0° face, the differences between adjacent faces are computed as

$$D(N) = \theta_N - \theta_{N+1}$$

where $N = 1, 2, \dots, 12$ and, for the twelve faces polygon, we have: $\theta_{12} = \theta_1$. Then, the angular error of each face referred to the nominal value is given by:

$$E(N) = D(N) - \frac{1}{n} \sum_{N=1}^n D(N),$$

where $n = 12$ for our case.

3. EXPERIMENTAL RESULTS

The experimental set up depicted in Fig. 1 was built. In order to avoid multiple secondary reflections, a pellicle beam splitter was used. Due to the minimum division of our autocollimator (0,5 arc seconds), several series of measurements were done to minimize the standard deviation and uncertainty values, this last one was of $\pm 1,0$ arc-sec. Table 1 shows the errors of a polygon obtained with the proposed method as well as the data obtained with the two autocollimators method.

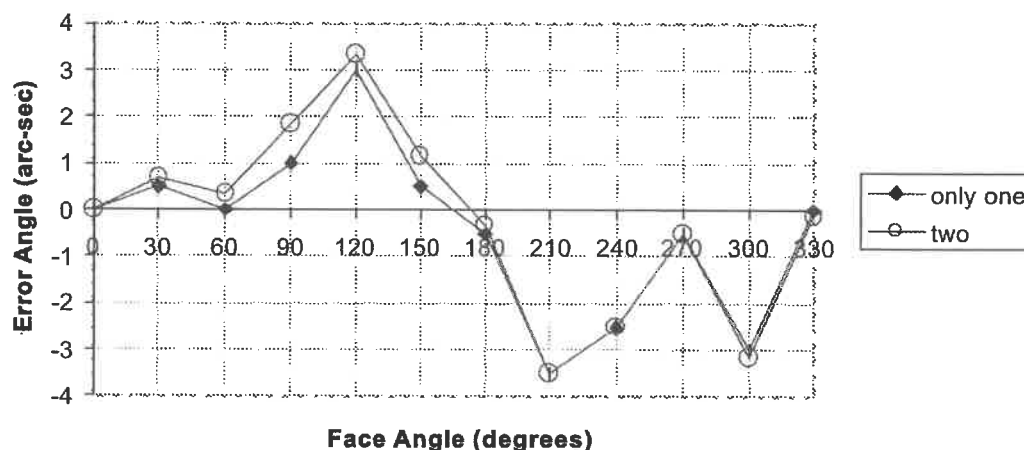
NOMINAL ANGLE [degrees]	ERROR (average) [arc-sec]	
	Only one	two
0	0	0,0
30	+0,7	+0,5
60	+0,3	+0,0
90	+1,8	+1,0
120	+3,3	+3,0
150	+1,2	+0,5
180	-0,3	-0,5
210	-3,5	-3,5
240	-2,5	-2,5
270	-0,5	-0,5
300	-3,2	-3,0
330	-0,1	0,0

Table 1. Comparative errors obtained from only one and two autocollimators

The uncertainty of two autocollimators method was $\pm 1,3$ arc-sec while the other was $\pm 1,0$ arc-sec. From Fig. 2 we can appreciate that the plots of the measured polygon errors are very similar.

4. DISCUSSION AND CONCLUSIONS

We have presented a method for measuring the error angles for one optical polygon with only one autocollimator. In comparison with the other methods, the experimental set up is very simple and reliable. Also, aligning and measuring take a little time. As an additional advantage, we use the same formulation described by Nava [4] to obtain the polygon errors. The experimental results obtained with the proposed method as well as those obtained by using two autocollimators were also presented confirming the performance of the method.



Graphic 1. Comparison of errors obtained from methods employing only one and two autocollimators.

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Optimal design of high stiffness XYZ 3-axis stage for AFM system

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Atomic force microscopes(AFM) provide high resolution, 3-D data and are therefore very useful for measurement of micro- and nano-structured objects. To establish of standard technique of nano-length measurement in 2D plane, we are developing AFM system. Because measurement uncertainty of this instrument is dominantly affected by the Abbe' s offset error of XYZ stage[1], XYZ flexure stage is designed to have least parasitic motion. In this paper with the object of reducing XYZ stage error, decoupled XYZ stage is suggested and optimal design is performed to maximize its robustness about disturbance. And setup of sensor system is done with the same object.(Fig. 1) 3-axis heterodyne interferometer which is commercial product of Zygo(ZMI 2000) measures the XYZ displacement. Each XYZ beam enters at the same place with the tip. For direct Z axis measurement 90 degree deflection prism below sample is used to deflect the laser beam. Using a prism, instead of a mirror, causes less polarization change. Soon to acquire high accuracy, COXI(Combined Optical and X-ray Interferometer) will substitute for commercial sensor.

For XY stage(Fig. 2(a)), single module parallel-kinnematic flexure stage is used which has high orthogonality and minimum out-of-plane motion. And Z stage(Fig. 2(b)) is proposed to have minimum out-of-plane motion also. Z stage is combined on XY stage. As a result XYZ stage has independent motion.

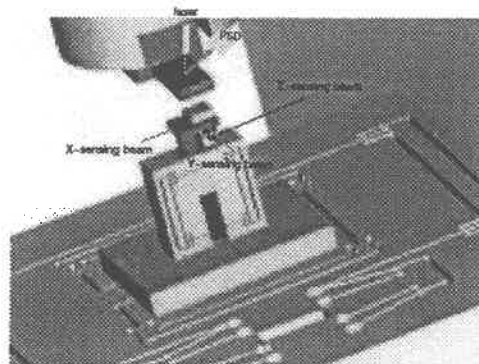


Fig. 1 The configuration of XY stage

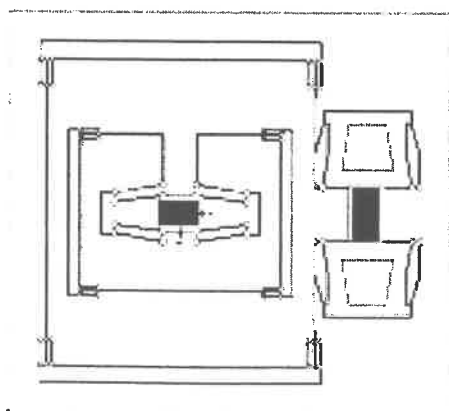


Fig. 2(a) The configuration of XY stage

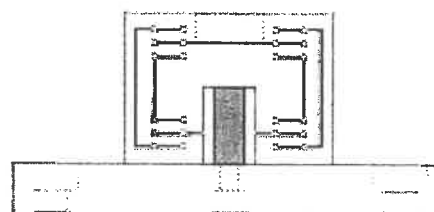


Fig. 2(b) The configuration of Z stage

In X stage magnification structure is used to magnify PZT's motion range (25 μ m). Y stage has same structure except it is inside X stage. Optimal design is performed to obtain best performance. Before modeling first we simplified model on the basis of FEM simulation to make more practical by reducing design variable. For example, the ratio between width and length of hinge's bar is determined by FEM simulation. Then the method introduced in Ryu's paper[2] is used to obtain dynamic equation. In Fig.3 flow chart of optimal design is shown. By this method the equation of motion is expressed by the function of design variables. Design variables are hinge thickness, hinge radius, distance between hinges and geometry data. The number of design variables 26.

For XY stage, to be robust about parasitic motion, optimal design of maximizing Z and tilt stiffness is performed under the constraint of motion range (110 μ m, 10% safety factor) and stage size (300mm*300mm). And for Z stage, optimal design of maximizing 1st resonant frequency is performed under the constraint of motion range (15 μ m, 50% safety factor) and stage size which enables locate on Y stage. If resonant frequency is get higher, scan speed is improved. So it makes reduce the error by sensor drift.

Design results are as follows. XYZ axis motion range is each 110 μ m, 110 μ m, 15 μ m. 1st natural frequency is 115Hz, 201Hz, 2.24kHz. And 6 axis stiffness is shown in Table. 1. From FEM simulation is similar with modeling results, we can verify modeling is reasonable. From FEM simulation, XYZ axis motion range is each 103 μ m, 105 μ m, 12 μ m and the 1st natural frequency is 109Hz, 201Hz, 2.86kHz. From that XY direction natural frequency is very similar but Z direction natural frequency has 46% difference. The reason is that to increase natural frequency in optimal design hinge's thickness is getting higher. So strain energy happens not only flexure hinge but also around flexure hinge (Fig. 4). But in the modeling only hinge stiffness is modeled and other connecting bars are assumed as rigid bodies. By that reason there is difference between modeling and FEM. But with simple model we can find that when increasing hinge's thickness its difference between modeling and FEM is increased linearly. By that fact optimal design is reasonable even if its result is different.

k_x (motion dir.)	2.3(N/um)
k_y (motion dir.)	3.2(N/um)
k_z	27.6(N/um)
$k_{\theta x}$	92.5(Nm/mrad)
$k_{\theta y}$	134.2(Nm/mrad)
$k_{\theta z}$	1018(Nm/mrad)

Table. 1(a) 6 axis stiffness of XY stage

k_x	2098(N/um)
k_y	470(N/um)
k_z (motion dir.)	47(N/um)
$k_{\theta x}$	81.9(Nm/mrad)
$k_{\theta y}$	374.6(Nm/mrad)
$k_{\theta z}$	255.3(Nm/mrad)

Table. 1(b) 6 axis stiffness of Z stage

With this optimal results, XYZ flexure stage is fabricated (Fig. 5). Its material is AL 7075.

To confirm XYZ flexure stage is well fabricated frequency response function (FRF) and maximum range test experiment is performed. As a result it has 1st natural frequency of 103.5Hz, 160Hz, 2.2kHz (Fig. 7(a)) and range of 106 μ m, 103 μ m, 11 μ m (Fig. 7(c)) in XYZ

direction each. Its experimental setup is shown in Fig. 6. And then by laser interferometer feedback(0.31nm resolution) we obtained 1nm stage resolution.

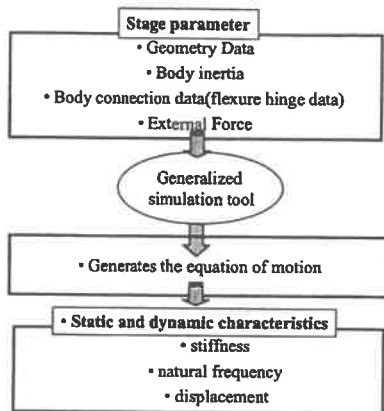


Fig. 3 Flow chart of optimal design

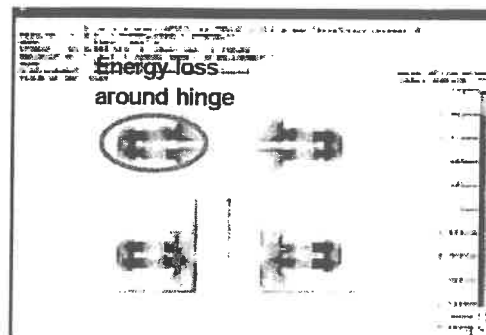


Fig. 4 Energy loss around flexure hinge of Z stage(FEM)

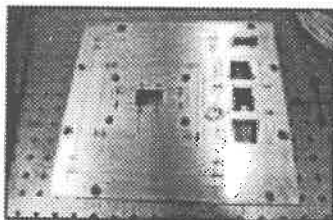


Fig. 5(a)

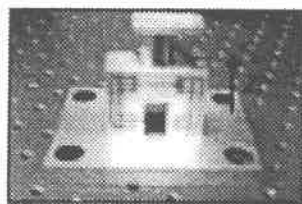


Fig. 5(b)

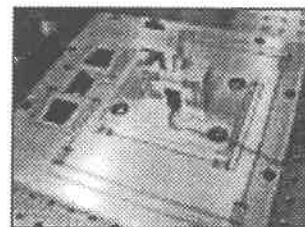


Fig. 5(c)

Fig. 5 The picture of XYZ stage : (a) XY stage, (b) Z stage, (c) XYZ stage

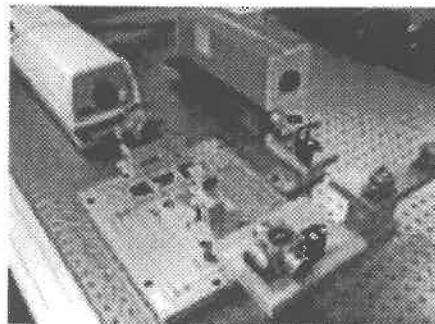


Fig. 6 Experimental setup of XYZ stage

Soon, total AFM system will be completed. Then total measurement uncertainty will be evaluated. By this stage we expect to reduce abbe' s offset erro because this stage has very low out-of-plane motion. Out final goal is to measure 2-D pitch with 3nm uncertainty about $100\text{ }\mu\text{m} \times 100\text{ }\mu\text{m}$.

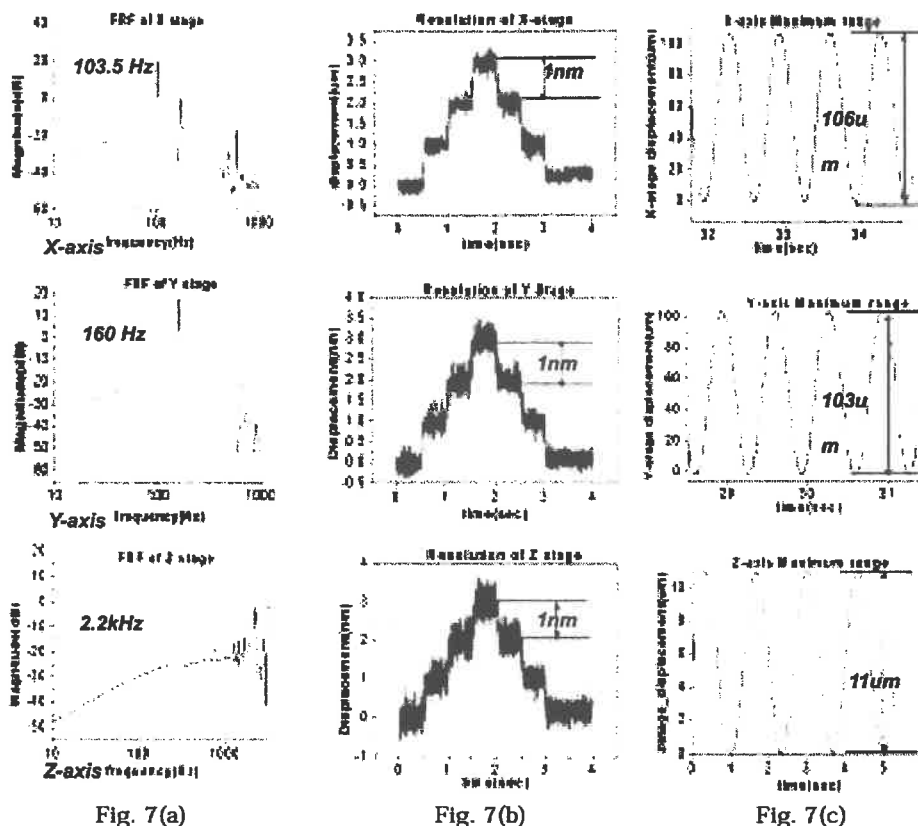


Fig. 7 Experimental results :

a) 1st natural frequency of XYZ stage b) step response c) maximum range of XYZ stage

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Artifacts and Test Procedures for Contour Measuring Instruments

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1 Introduction

Contour measuring instruments [1] are applied in mechanical engineering and optics industry. They are running both, in the workshop and in air-conditioned measurement rooms. This is reflected in a variety of test objects and measurement tasks. Amongst these are outer and inner contours, where radii, angles and sizes have to be determined. Typical tolerances of 0.05 mm and less are a challenge for the instruments. The ability of the instruments for such tasks has to be checked by acceptance tests and in the framework of quality management by periodic interim checks and reverification tests. Prerequisites are suitable embodied standards, unified parameters, and test procedures. For contour measuring instruments these prerequisites are not fulfilled as yet. Therefore, three major manufacturers of contour measuring instruments and the PTB have run a joint project to fill these gaps. Suitable artifacts and a test procedure were developed and investigated. One artifact was developed to be used as a working standard for testing contour measuring instruments and to be calibrated by a calibration laboratory. Moreover, another more precise artifact was developed within the project to be used as a transfer standard closing the traceability chain from the calibration laboratory to the NMI (e.g. PTB).

2 Working Standard

The working standard is made from stainless steel by electrical discharge machining (EDM). It is robust, easy-to-use, and relatively cost efficient. Therefore, virtually any contour measuring instrument may be equipped with it.

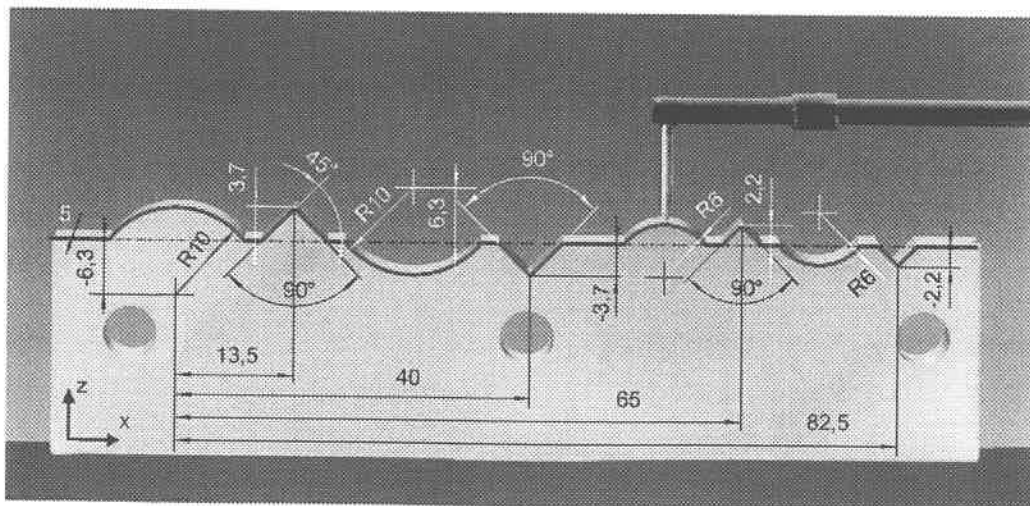


Figure 1: Working standard.

The dimensions of the working standard match the typical measurable range of most contour measuring instruments and such all parameters may be checked by a single measurement. For the first time this working standard provides a combination of convex and concave geometric elements with low form deviations. The concave elements are symmetrical to the convex ones. The dimensions differ only by some micrometers from perfect mirror symmetry. This is especially important for instruments applied for workpieces like bearing bushings or valve seats. The surface is very homogeneous and reflects diffuse. The surface roughness amounts to about $R_z \approx 1 \mu\text{m}$. The working standard is, therefore, excellent suited to test optical measurement systems too, e.g. fringe projection systems.

In principle, the working standard can be calibrated using Scanning-CMMs or contour measuring instruments. Until now, no calibration laboratory is accredited for this task. Therefore, the PTB developed a special calibration method (Patent pending). This method is based on a vicariousness optical calibration of auxiliary artifacts as described as follows. To reduce the manufacturing effort, the working standards are manufactured in stacks. Between the semifinished plates of a stack 0.2-mm sheets are placed. After manufacturing with EDM, all parts of a stack have the same contour. The sheets serve as auxiliary artifacts which are calibrated with an optical CMM (cf. Figure 2). The systematic errors of the CMM used are compensated by applying the substitution method according to ISO 15530-3 [2]. The reference standard is a calibrated 2D-Chrome standard having the same contour like the sheets and the working standards respectively (cf. Figure 3). Moreover, this 2D-Chrome standard is used to optimize different measuring parameters, e.g. the backlight illumination brightness (cf. Figure 4).

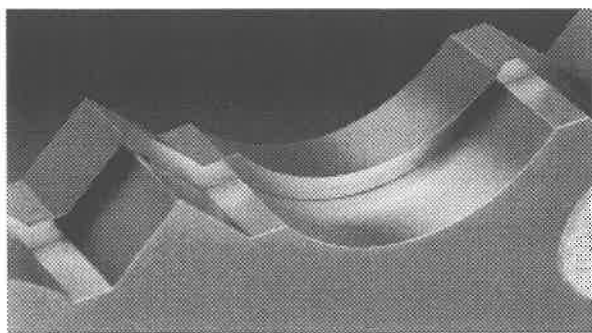


Figure 2: Sheet metal 0.2 mm, fixed between two working standards for optical calibration (Detail).

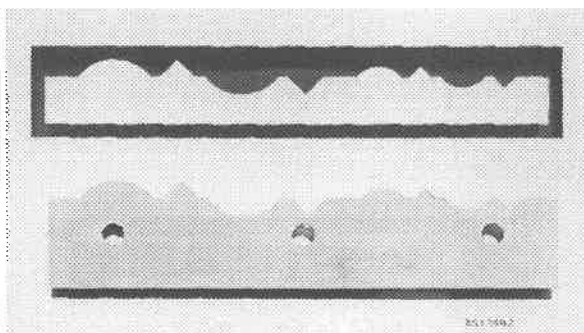
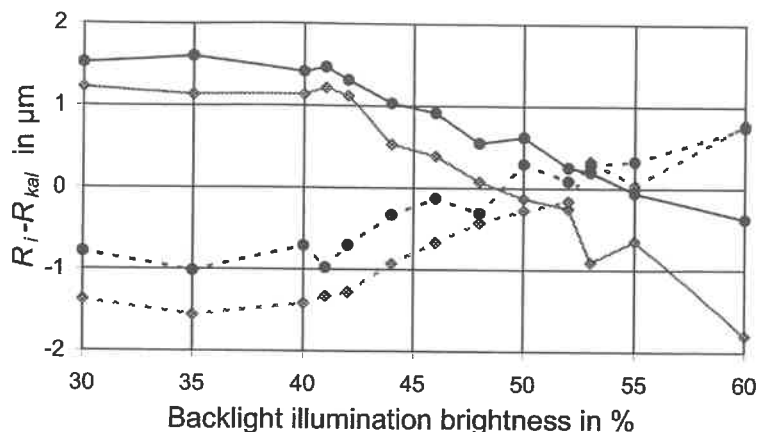


Figure 3: 2D-Chrome standard (above) and sheet metal (below) for substitution method.

The method used to calibrate the working standards allows to realize uncertainties for all measurands within $1 \mu\text{m}$ or 0.05° . The largest uncertainty contribution is caused by the objects themselves and is determined, firstly, by investigation of the fully manufactured stack and, secondly, by comparison of all the sheets of a stack. The uncertainties for the new optical calibration method were verified by comparison measurements with an Scanning-CMM.

Figure 4: Determination of the optimum illumination brightness at convex (—) and concave (- - -) circular arc segments, ●: $R = 10$ mm, ◆: $R = 6$ mm.



3 Transfer standard

The transfer standard is composed from single elements with very low form deviations: a length standard, two spheres and a prism wrung to a flat. Using single elements allows an individual calibration with the lowest available uncertainty. The elements are fixed at different modules, which are pre-adjusted with respect to a common base-plate and, therefore, can be quickly exchanged without time-consuming adjustment (cf. Figure 5 and 6). The modularity allows an easy extension to new measurands.

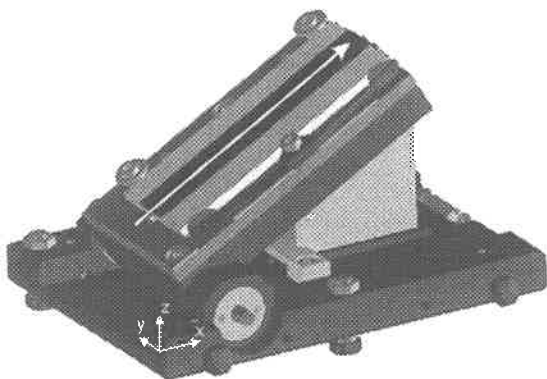


Figure 5: Transfer standard: length module.

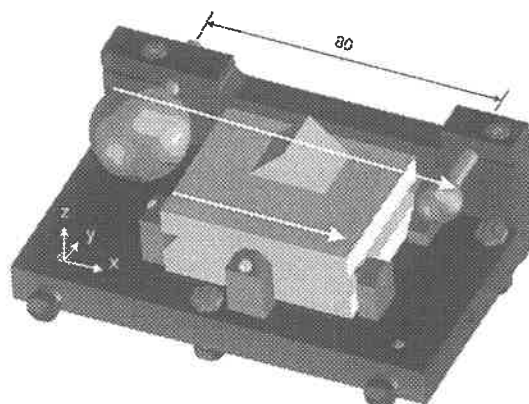


Figure 6: Transfer standard: module for radii, angle and straightness.

The heart of the transfer standard is a new kind of length standard made from silicon, which may be measured in different inclinations (cf. Figure 7).

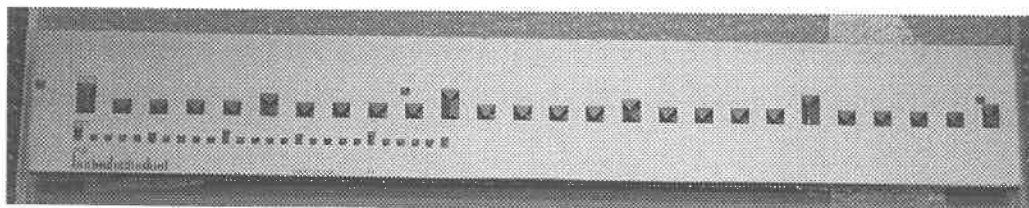


Figure 7: Length module: silicon length standard.

Using this new length standard contour measuring instruments can be tested in analogy to the test of CMMs with the aid of step gauges. The calibration of the different measurands of the transfer standard is performed by suitable standard measuring equipment at PTB. The achieved uncertainties for distances and radii remain under 0.1 μm , for straightness deviations under 0.02 μm , and for angles under 0.001°.

4 Test procedure

The results of the project are used to develop a draft for a guideline entitled "Accuracy of contour measuring instruments. Parameters and their reverification. Part 1: Acceptance and reverification tests for contour measuring instruments using contact profile method." The test procedure is basically related to ISO 10360-2 and 10360-4 [3, 4]. This guideline is being discussed in a VDI/VDE-committee (German Association of Engineers).

5 Outlook

Until the end of 2003 comparison measurements are running both with tactile and optical methods. The aims of these comparison measurements are:

1. Determination of the state of the art of the measurement of geometrical standards (3D and 2D) with relatively complex contour.
2. Use of the results with respect to the accreditation of calibration laboratories within DKD.
3. Assignment of the experiences made for the further development of the "Virtual CMM" in scanning mode [5].

The tactile measurements are carried out at two working standards (Figure 1) by 13 participants using 9 Scanning-CMMs from 4 manufacturers¹ and 15 contour measuring instruments from 8 manufacturers². The optical measurements are executed at the 2D-chrome standard (Figure 3) by 7 participants using 9 optical CMMs from 7 manufacturers³. The results of the comparison measurements will be published presumably in spring 2004.

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¹ Leitz, Mitutoyo, Werth, Zeiss

² Hommel, Mahr, Mitutoyo, Optacom, Semicronic, Taylor Hobson, TSK, T&S

³ Mahr Multisensor, Mycrona, Mitutoyo, OGP, OKM, Wegu, Werth

Development of Ultra precise Gear Measuring Instrument

First report: Evaluation on the Accuracy of Principal Axis

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1. Introduction

High precision gears, with geometrical accuracy of sub-microns in the field of motion control machine systems such as high precise industrial robots, are steadily becoming required for their accurate operation and for their smaller sizing demands by means of finer module gears and less-back lash system.

In order to produce such high precise gears, every tools and machines used in the process must be ultra precise. At the same time the measuring instrument with accuracy of one order higher than the quality of finished products should be used.

However, the reality of attainable accuracy of existing measuring instruments in the market is at most around 3~5 [μm].

The authors keenly felt that an ultra-precise measuring machine should be developed urgently for the purpose of research, experimental make and actual production as well.

The obtained result on the accuracy of the principal axis movement, will be described within the first report of this project.

2. The Target of the Accuracy on the Measuring Machine

The accuracy target with this Measuring Machine

We plan to construct an Ultra-precise three dimensional gear measuring machine with pure mechanical open system to measure the individual deviations (tooth profile, tooth lead, pitch, run-out etc.), also to eliminate the measuring errors and to improve the accuracy less than 0.1~0.3 [μm] in linear deviation and less than 1 [arc-sec'] in index deviation.

3. The Mechanism and Specifications

The machine consists of the two dimensional (X, Y) rectangular traveling table attached with measuring probe, and the traveling axis in vertical (Z) together with rotational (θ) motion for gear to be measured. The most remarkable point of the structure with this measuring instrument is that the indexing (θ) axis has vertical (Z) axis up and down motion.

The positioning mechanism for each axis in three rectangular directions (X, Y, Z) are using high precise trapezoidal lead screws, thread with 20°in (half) angle, which are driven by pulse-motors through Harmonic-drive

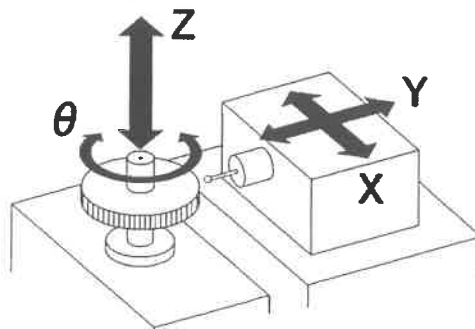


Fig. 1. Schematic view of mechanism.

reduction units.

The strokes along X, Y and Z directions are 160, 150 and 130 mm respectively.

The X-axis is guided with parallel guide bars of different diameter, having a pair of cylinder type air bearings for each guide bars. And the Y-axis is guided with a pair of parallel V-groove slides and two pair of cylinder type air bearings on each side for the table weight balancing.

The Z-axis is linearly guided with two pairs of cylinder type air bearings with different diameter, the upper pair is guide for Z-axis motion and indexing θ -rotation and the lower pair is for linear Z-motion guide only. For the purpose of indexing torque bearings a pair of additional guide ways along with Z-axis are equipped with a cylinder type air bearings for each guide bars.

The center support for the gear to be measured on upper portion of the machine is linearly guided with a pair of cylinder type air bearings.

The θ -rotation axis is guided with the cylinder type air bearing which shares its function with Z-axis guide, and is equipped with ultra precise axial type air bearing at the connecting area between upper Z- θ -axis and lower Z-feed mechanism.

The ultra-precise internal gear indexing system is installed between these axes and is driven by pulse-motor. Additional rotary-encoder of 0.4 [arc-sec] guaranteed accuracy is installed between these axes for comparison measurement of the indexing error.

The differential Capacitance type two dimensional micro-displacement measuring probe is used as the sensor. The resolutions of the sensor are 10 [nm] in X-axis direction and 50 [nm] in Y-axis direction respectively. The measured data of the deviations is digitalized and stored into personal computer and analyzed to obtain required values.

The machine is settled in a climate controlled room at $23 \pm 0.1^\circ\text{C}$ in temperature and less than 35% in humidity in order to eliminate unreliable deviations as due to heat expansion.

4. The Positional Deviations of Movement along the Guides and the measurement method of the Deviation.

For obtaining the positional deviations within 0.1~0.3 [μm], the positional deviations along X,Y,Z and θ -axis should be kept within 0.1 [μm]. Especially, the deviation on the most important axis "principal axis (Z, θ)" should be less than the above mentioned value. We aimed to keep value less than 1/10 of the mentioned value, therefore the aimed value becomes less than 10 [nm] when the axis is turning.

a) Measurement of run-out in horizontal plane.

Precisely finished inner surface of a cylinder is used as the standard surface of the measurement and the disc with 12 electrodes on the outer cylindrical surface is used as the capacitance sensor carriage. The gap between the electrodes and the standard surface is about 10 [μm] which

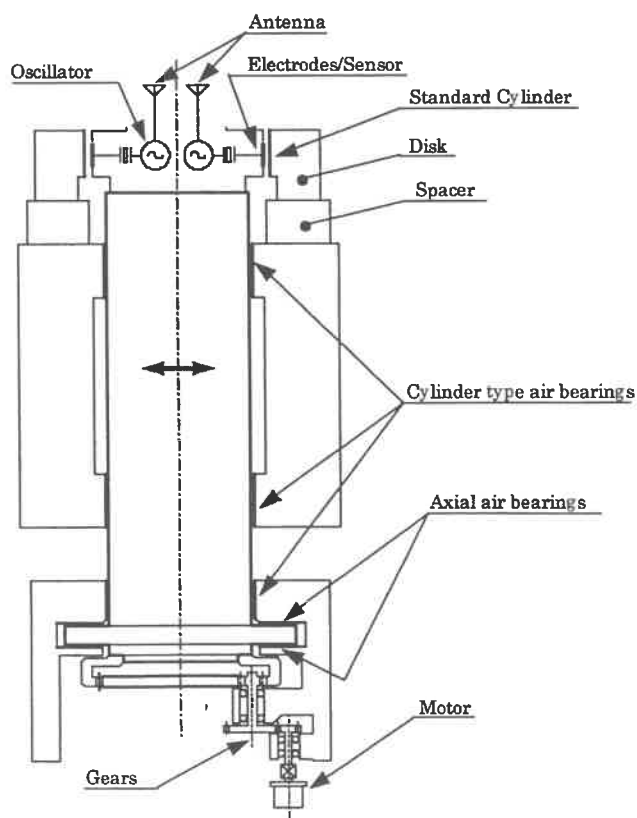


Fig. 2. Outline diagram of principal axis.

gives aimed resolution of the deviation. By turning the sensor carriage and measured signals are wireless transmitted to the computer. The measured values : $F_i(\theta)$ are expressed by the following equations.

$$F_i(\theta) = r(\theta) \cdot \cos\{\theta - \psi(\theta) + \phi_i\} + Q_i(\theta + \phi_i)$$

$$G_i(\theta) = K_i \cdot F_i(\theta)$$

There, $r(\theta)$: run-out amplitude of principal axis
 $\psi(\theta)$: phase angle from X axis in counter clockwise direction
 i : number of sensor,
 θ : rotational position of principal axis,
 ϕ_i : angle between the first and i-th electrode,
 K_i : magnification factor of i-th electrode,
 $Q_i(\theta)$: run-out amplitude of standard surface,

By self compensation processing, the effect of geometrical deviation on the standard cylinder can be neglected and the values of the run-out are obtained through Fourier analysis with 12 phases of data.

b) Measurement of the deviation in vertical direction.

The capacitance type micro-displacement measuring probe is set on the top of the principal axis and the measured data is sent to the computer.

5. Some examples of measurements

a) Figure 3 shows the measured result of the horizontal run-out on the principal axis. Ten diagrams of datum are shown on one chart and it can be seen that the maximum value of the amplitude is less than 20 [nm].

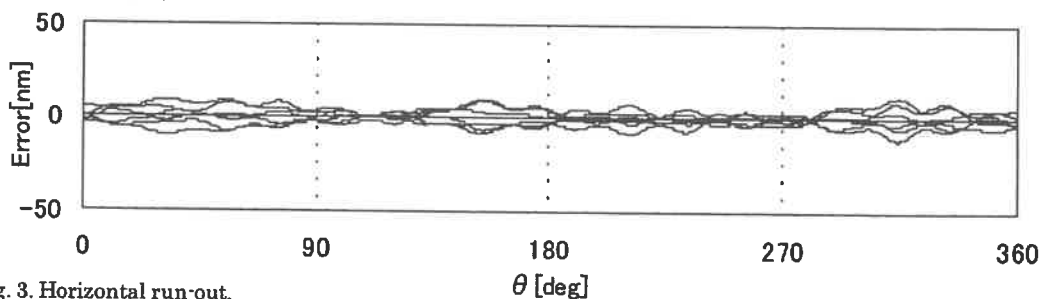


Fig. 3. Horizontal run-out.

b) Figure 4 shows the measured result of the vertical run-out on the principal axis. It can be seen also that the maximum value of the amplitude is less than 20 [nm].

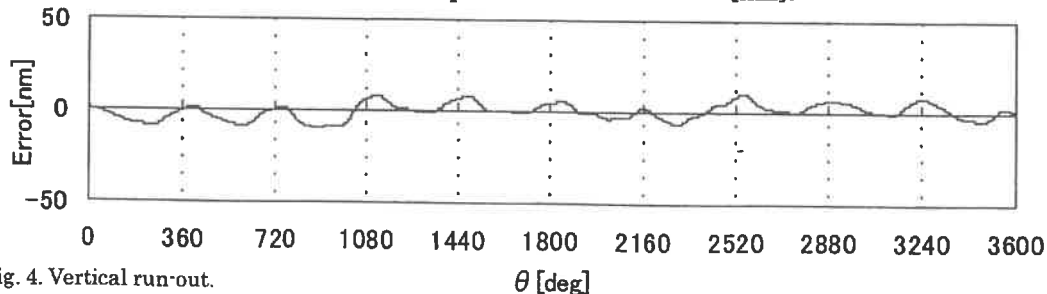


Fig. 4. Vertical run-out.

6. Discussion

- a) From the Fourier analyzed results of the measured datum on the horizontal run-out of the principal axis, 15th and 17th harmonic components in one revolution were detected with amplitude of around 3 [nm]. We are discussing about the reasons of these deviations. However, the amounts of these values are negligibly small. It is suspected that the 16th and 18th harmonics can be detected since the reduction ratio with the two stages of the gear train in this indexing system are 1/9 and 1/16. However, these were not detected actually. The unstable waving was recognized from the chart in continuous driving of the principal axis. It seems to be the result of the variation of air pressure for the air bearings, but was because of the small restriction on the way out of discharging air.
- b) The measured value of axial run-out was less than 20 [nm].
And there is one more step to achieve to the aimed value of 10 [nm]. Also, it seems to be caused by the same reason discussed above.

7. Conclusion

By combining the above discussed result with our past obtained technical result, we can foresee realization of the Ultra-precise Gear Tester.

Laser Tracker Compensation Using Displacement Interferometry

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Introduction

Laser trackers have become the tool of choice for large-scale coordinate measuring needs. The requirement to rapidly validate the performance of these instruments to ensure the integrity of their measurement results is critical. The task is complicated by the fact that the measuring envelope of these instruments is quite large (> 35 meters). Such large measurement volumes often require the use of large length standards, e.g., greater than two meters, to characterize the instrument performance. Physical standards such as large calibrated artifacts, commonly referred to as scale bars, may suffer the problems of being unwieldy to use, difficult to construct and quite often costly to maintain.

In response to the need to characterize the performance of this class of coordinate measuring system, NIST, with funding from the U.S. Navy Calibration Coordination Group (CCG), developed the Laser Rail Calibration System (LARCS). The system includes a linear interferometer and support structure that can be configured to support a three meter calibrated length in any arbitrary position inside the tracker work-volume [1]. The results of displacement measurement tests performed with the LARCS contain valuable information about the laser tracker performance characteristics. However, there are very few tools available to exploit the vast information contained in these data [2][3].

The primary motivation for the investigation provided in this paper, is to provide users of these systems with an example of a systematic approach for utilizing displacement measurement for characterizing the performance of laser trackers. Estimates of the error model parameter obtained from displacement measurements can be used either to compensate the laser tracker or, along with error propagation tools, such as Monte Carlo Simulations, to estimate the point coordinate uncertainty. For this paper we will show that estimates of the error model parameters can be obtained using linear displacement measurements.

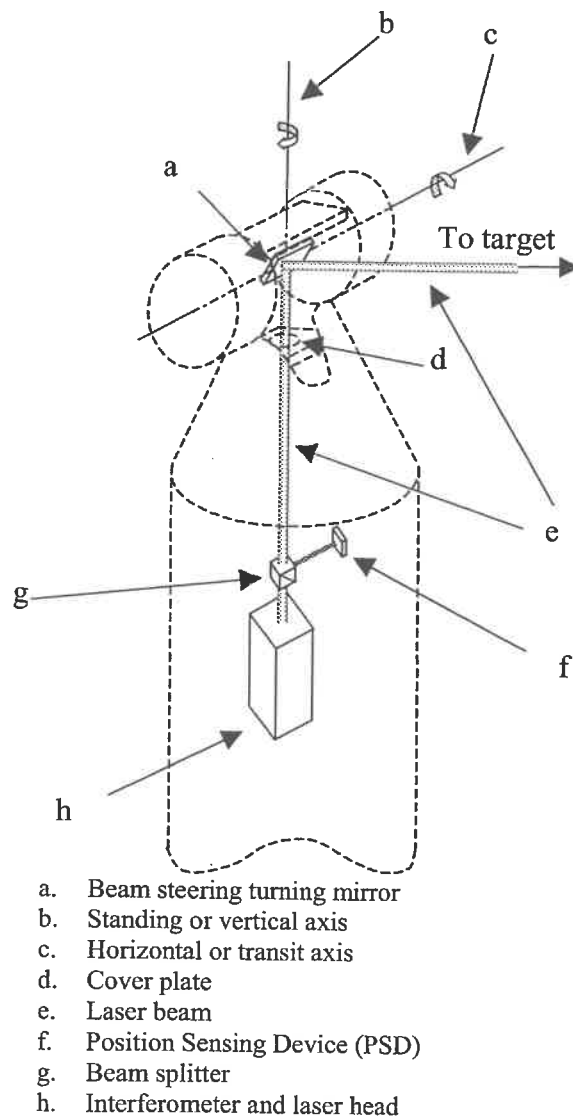


Figure 1. Laser Tracker Schematic

Background

The construction of laser trackers, similar to a theodolite, makes it possible to measure to a fixed target using two different instrument configurations. That is, the tracker can track to a fixed location measure a target, then the tracking mirror (**Figure 1**) can be rotated 180 degrees about an axis pointing vertically and then rotated about a horizontal axis to point back at the target. Measurements performed in this way are referred to as two-face or front sight/back sight measurements. These measurements are important because many of the tracker error model parameters are reversed and are thus accentuated using this procedure. (The error model parameters represent physical position and orientation errors of the optical components.)

Because of the error reversal property of two-face measurements, manufacturers often employ this technique to calculate estimates of the error model parameters. Sets of two-face measurements are performed and estimates of the model parameters are obtained using a least squares algorithm. These measurements however do not encompass a large portion of the model parameters that do not reverse and, thus, are not accentuated by this technique. To estimate these parameters, additional measurements are often employed. These obligatory measurements typically utilize an un-calibrated fixed length artifact, a rotating ball bar, so that at no time during the compensation is scale introduced into the calibration procedures. Such procedures are frequently referred to as self-calibration techniques or closure techniques and are perfectly valid [4].

Although closure techniques are a very powerful tool the use of a calibrated reference artifact for tracker compensation can be advantageous. The benefit of using an independently calibrated interferometer system is that all measured lengths are traceable to the SI unit of length, *i.e.*, the meter. Additionally, measurement positions can be selected so that they are sensitive to all of the tracker error model parameters.

Compensation Algorithm

The LARCS system is designed to establish and position a reference length in any arbitrary orientation and location inside the tracker work-volume. Although the software provided with the LARCS has provisions for performing two-face measurements, the most common use for the system is to compare the reference length to the values for the same length measured by the laser tracker. The output of a measurement is the distance and horizontal and vertical angles to each point that comprise a length measurement, as well as the calibrated value for the lengths.

Ideally the calibrated value and the measured value for the length would be equal. However, because of the presence of the geometry errors in the construction of the tracker, other systematic errors not included in the model and random effects this is almost never the case. This measurement error which is also calculated and stored by the LARCS software is the distance as measured by the laser tracker minus the calibrated value for the length or equivalently:

$$\varepsilon = D(\bar{Y}, \bar{X}) - D_{calib} \dots\dots\dots(1)$$

Where $D(\bar{Y}, \bar{X})$ is the distance function in spherical coordinates, Y is a vector of the measured coordinates and X is a vector containing the unknown alignment parameters. We can write the function in this form because the corrected coordinates are a function of the measured coordinates and the unknown alignment parameters. If multiple lengths are performed Equation

(1), after linearization, can be expressed in matrix notation by the following equation:

$$\bar{\varepsilon} = J(\bar{X}) + v \dots \dots \dots (2)$$

Where $\bar{\varepsilon}$ is the vector of measured errors, $\bar{X}$ is the vector of unknown alignment parameters, J is the Jacobian and v is a vector of residuals [5]. This equation has the familiar form $AX=b+v$ and can be solved easily using standard mathematical techniques [5][6][7].

Procedures and Results

The above algorithm was programmed in MATLAB. The simulation generates a random set of alignment parameters. The error model is used along with a set of randomly generated point pairs to create a set of displacement measurement errors (simulated measurement errors). These errors are used in the least squares fit algorithm described above to test the correct implementation of the model. Additionally, the simulation provides an invaluable tool for selecting measurement positions that are sensitive to a significant portion of the alignment parameters. That is, it is not an easy task to determine in which positions and orientations the reference length should be measured. By generating random measurement positions and calculating the model parameter sensitivity coefficients, the placement of the reference length can be judiciously selected to be sensitive to the largest number of the model parameters. Analysis of variances is employed and model parameter confidence intervals calculated to determine the number of requisite measurement positions. Obviously as the number of measurement positions increases the confidence intervals for the model parameters get narrower. For this test, only 54 unique measurement positions were employed this is a compromise between an acceptable level of confidence and the available time to perform measurements. **Figure 2** shows a frequency of the residual length measurement errors before and after compensation. The standard deviation of the residual length errors were decreased by a factor of two.

After obtaining estimates of the error model parameters, a simple test was performed. The tests consisted of measuring 12 targets rigidly mounted to a thermally stable structure. The position of each of the target nest was measured with the tracker placed in 4 different locations and orientations relative to the support structure. Each set of twelve measurements was fit together in a rigid body least squares sense. The deviation from the average calculated position for each target nest is plotted for both the compensated and uncompensated points. Chart 1

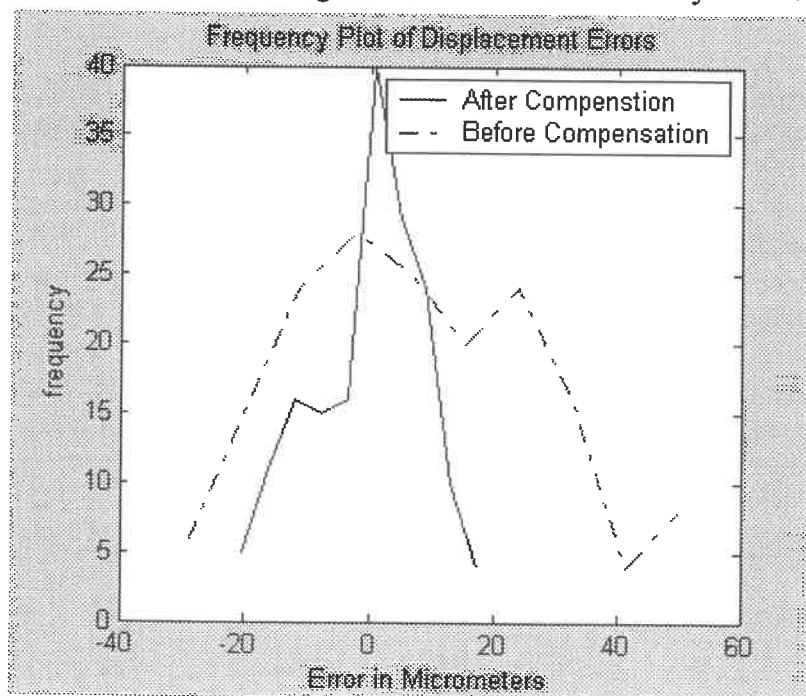


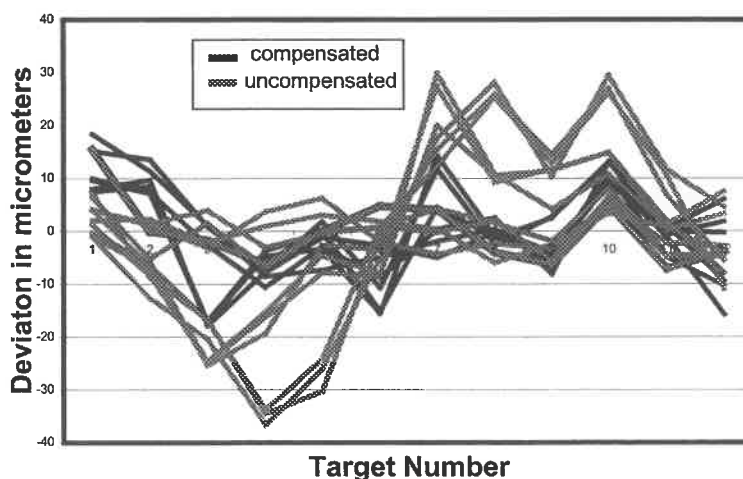
Figure 2. Residual length errors before and after correction

implies that a significant reduction in the systematic variation in the measured points can be obtained using this methodology. It is important to note that the deviations shown in the chart are within the manufacturer's specification yet significant systematic variations still exist. This variation may, however, be due to poor calibration of the laser tracker. At the time of this writing the laser tracker is being compensated again using the manufacturer's recommended practice. A set of new model parameters will again be calculated using the methods described here. A similar analysis will be performed to determine if in fact the residual point variation is caused by operator error or lack of sensitivity of the recommended compensation procedures to all of the error model parameters.

Conclusion

Length measurement results show that displacement measurements can be used to obtain estimates of laser tracker error model parameters. Experimental measurements imply that displacement methods can significantly reduce the variation in the point coordinate variation by nearly one half. When used with simulation tools such as Monte Carlo simulation software, this methodology might in the future be used to calculate point coordinate uncertainty for any measured coordinate.

Chart 1. Deviaton From Mean Value



Work will continue to apply this methodology to other similar trackers. It is our hope that this method can be extended for use with trackers with other kinematic configurations as well.

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Tip Waviness Compensation in a Polar Profilometer

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In 2001, the North Carolina State University Precision Engineering Center unveiled the polar profilometer *Polaris* [1]. *Polaris* was based on a system of two stacked axes with a linear axis residing atop a rotary axis. Using this system, an LVDT probe could be positioned to allow measurements of high aspect ratio aspheric surfaces not possible with other instruments. Initial specifications were for an overall measurement accuracy of 500 nm over a 50 mm circular measurement field. To achieve this level of performance, geometric machine errors such as probe tip misalignment, linear axis pitch and probe radius had to be compensated. As *Polaris* has given way to its commercial sibling, *Ultraform 2D*, shown in Figure 1, accuracy requirements have risen and perpetuated the need for more error compensation. Foremost, and most challenging of these new errors to address is probe tip waviness compensation.

The three limiting criteria presented for probe tip waviness on *Ultraform 2D* are: a commercially available Grade 5 (± 125 nm) probe tip must be used, waviness errors must be measured expediently on the instrument, and compensation must be performed automatically. Thus, achieving overall accuracy better than 100 nm requires more than simply choosing a more accurate and expensive probe tip.

The process of waviness compensation requires that, first, the waviness of the probe tip be evaluated and, second, the result be applied to every subsequent measurement. The unique geometry associated with a polar measuring system allows a certain degree of self calibration. This feature of the profilometer was exploited in *Polaris* to align the instrument's measuring probe with the rotary axis, thus establishing a true polar coordinate system. Two orthogonal alignment errors were identified: radial offset (ρ_0), and tangential offset (τ_0). Radial offset has a straightforward effect on a measurement in that it merely needs to be added to the measured radial

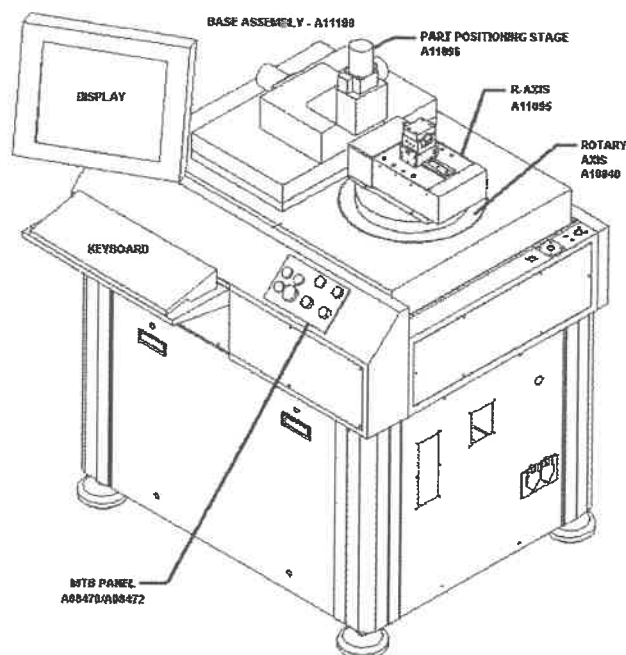


Figure 1. The *Ultraform 2D* polar profilometer.

position R in order to obtain the corrected position R' .

$$R' = R + \rho_0. \quad (1)$$

The impact on a measurement of the tangential offset is a bit more involved, since its influence is expressed as a function of both R and θ . Hence, the corrected radial position R'' is

$$R'' = (R'^2 + \tau_0^2)^{1/2}. \quad (2)$$

While the corrected angular position, θ' , is

$$\theta = \theta' + \tan^{-1} \left(\frac{\tau_0}{R'} \right). \quad (3)$$

Waviness (w), on the other hand, is a function of the probe contact angle (γ), which is itself a function of R and θ .

As shown in Figure 2, radial offset error (ρ_0), tangential offset error (τ_0) and waviness error ($w(\gamma)$) are all present in any measurement.

Waviness Evaluation

Evaluating the waviness of a given probe tip is inexorably linked to evaluation of probe misalignment. Probe misalignment is, in fact, just a special case of waviness with a high degree of symmetry. Unfortunately, this does not alleviate the need to find the relative position of the probe tip with respect to the rotary stage axis of rotation. The method for evaluating probe tip alignment that has been examined uses a flat as a reference standard.

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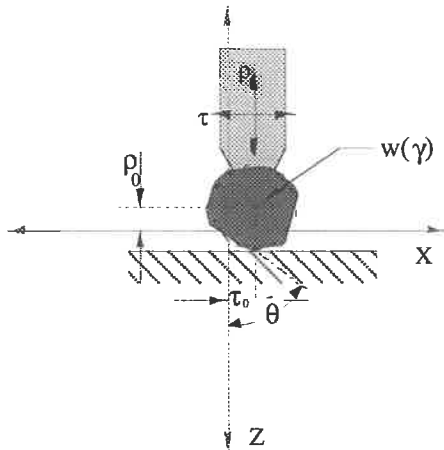


Figure 2. Probe error is a combination of radial offset error (ρ_0), tangential offset error (τ_0) and waviness error ($w(\gamma)$).

For a probe tip with a radius of r , the probe position will be $-r$ for all positions of θ if the center of the tip is positioned with its center at $\rho = 0$. One means of achieving this is by placing a flat plate ($\lambda/20$) at $-r$ as shown in Figure 3. If the probe is truly centered, the probe output will give only the probe waviness as the rotary axis is traversed through the angular extent of the probe tip. If, however, there is an offset in either the ρ or τ directions, the probe output will change in a characteristic way for each direction that is a linear combination of all these errors.

As shown in figure 4, or an offset of ρ_0 in the ρ -direction, the output of the LVDT will change as a function of the rotary axis position θ :

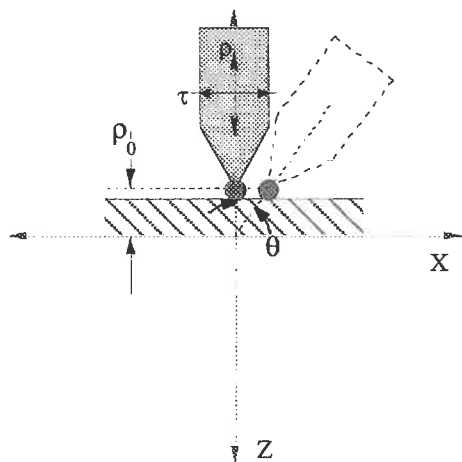


Figure 4. Schematic and plot of LVDT deflection with an offset of ρ_0 in the ρ -direction. The response as a function of θ can be used to determine the offset and adjust the machine.

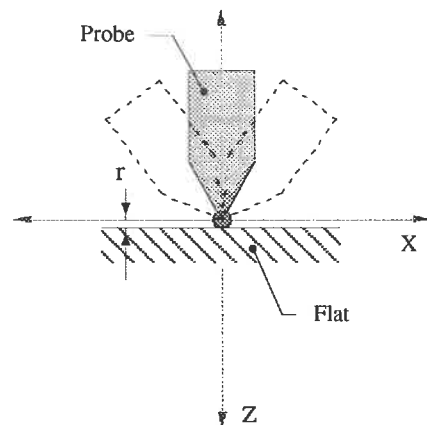


Figure 3. When the probe tip's center is aligned with the machine origin, rotation of the θ -axis will yield only waviness data.

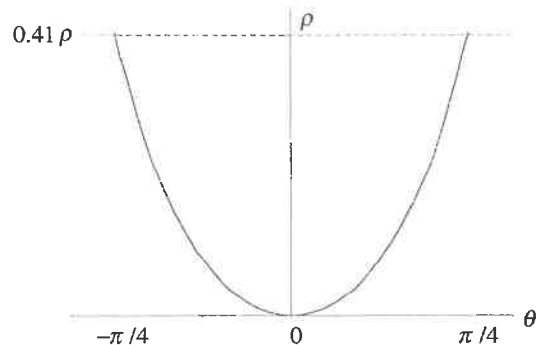
$$\rho = \rho_0 \left(\frac{1}{\cos \theta} - 1 \right). \quad (4)$$

When an origin measurement is made and the ρ -offset is measured, the R -axis zero can be reset to eliminate the offset. In addition to ρ -offsets, there can be an offset in the τ -direction. This offset produces a different response when measured using the flat method. The response of the LVDT to a τ -offset is a function of θ as shown in Figure 5 can be written:

$$\rho = \tau_0 (\tan \theta). \quad (5)$$

Again, this characteristic output can be used to detect and correct the offset – this time in the τ -direction.

In the actual alignment procedure, the



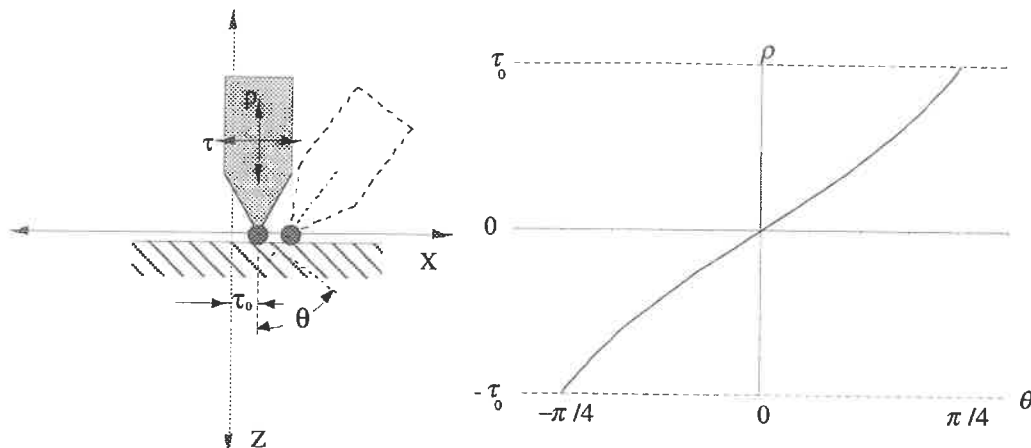


Figure 5. Schematic and plot of LVDT deflection with an offset of τ_0 in the τ -direction.

situation is slightly more complex. A combination of offsets will produce an output of the probe according to the sum of Equations (4) and (5),

$$\rho = \rho_0 \left(\frac{1}{\cos \theta} - 1 \right) + \tau_0 (\tan \theta). \quad (6)$$

Armed with this knowledge, one can now collect data and perform a curve fit of Equation (6) to the collected data and simultaneously solve for ρ_0 and τ_0 . The residual error is the waviness of the probe since, for a flat, $\theta = \gamma$.

The disadvantage of this method is that, for large amounts of waviness, the determination of ρ_0 and τ_0 via a curve fit can be affected, increasing uncertainty in the location of the rotary axis center. Fortunately, by limiting

magnitude of the waviness to ± 125 nm, the influence on the fit is not appreciable.

Error Correction

Once the tip waviness and probe alignment errors are known, the data from subsequent measurements can be corrected in software. None of the errors are of sufficient magnitude to require real-time alteration of probe position during a measurement. Gage misalignment and nonlinearity are easily corrected by applying their compensation formulas sequentially.

Once the misalignment errors have been compensated using Equations 1-3, the procedure for probe radius and waviness correction is similar to that of modifying a tool path prior to a machining process. An important difference for a metrology instrument is that for each point in the

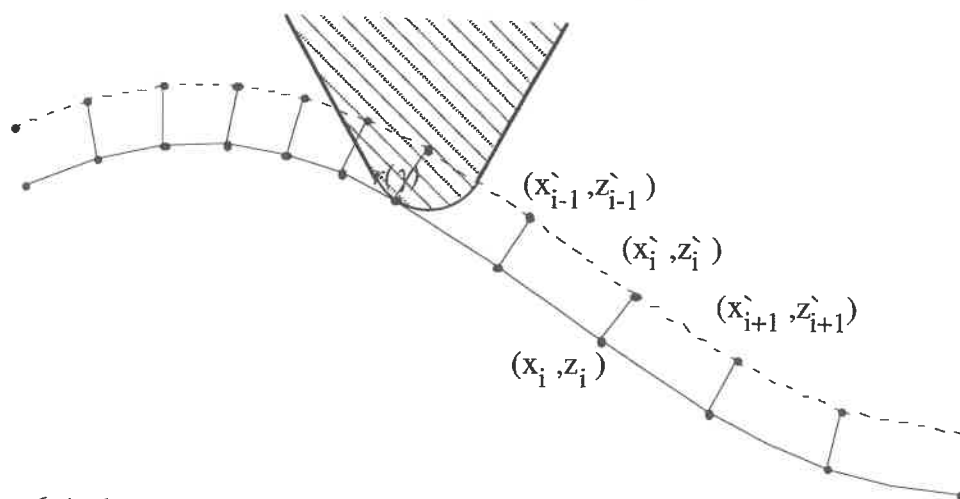


Figure 6. As the probe tip traverses the surface, the contact angle changes. This contact angle can be determined from the slopes of the interpolated line segments between data points.

data file the direction cosines of the correction vector must be estimated from the data itself.

In its most basic form, the waviness compensation is numerical and requires a minimum of three data points. The first step is the calculation of the slope of the surface at a data point (x_i, z_i) . Since the slope of a point cannot be calculated, two neighboring points assumed to be on the surface are used. The slope m between any two points is given by

$$m_i = \frac{x_i - x_{i+1}}{z_i - z_{i+1}}. \quad (7)$$

To desensitize the data somewhat to local fluctuations, the slopes of the neighboring points are averaged

$$\overline{m}_i = \frac{m_i + m_{i+1}}{2} = \frac{1}{2} \left(\frac{x_i - x_{i+1}}{z_i - z_{i+1}} + \frac{x_{i+1} - x_{i+2}}{z_{i+1} - z_{i+2}} \right) \quad (8)$$

The inverse of this point slope is the normal slope, which, when inserted into the point-slope form of the normal line

$$z_i - z_i = \overline{m}_i (x_i - x_i) \quad (9)$$

can be used to solve for each point at a distance of the probe radius r along that line.

Considering waviness as a polar variation in probe radius, a combined correction for probe radius and waviness is found using Equations (10) and (11):

$$x_i = x_i + r(\gamma) \cdot \overline{m} / \sqrt{\overline{m}^2 + 1} \quad (10)$$

Solving for z_i ,

$$z_i = z_i + r(\gamma) \cdot -1 / \sqrt{\overline{m}^2 + 1} \quad (11)$$

The function $r(\gamma)$ is the radius as a function of contact angle or the sum of the nominal probe radius r and the waviness $w(\gamma)$.

$$r(\gamma) = r + w(\gamma). \quad (12)$$

The contact angle γ is a function of the local slope m and the probe angular position θ :

$$\gamma = \theta - \tan^{-1} \left(-\frac{1}{m} \right). \quad (13)$$

The special cases of zero or infinite slope are easily detected by adding $r(\gamma)$ to x_i and z_i respectively.

For smooth surfaces with slowly varying slopes this process is relatively straightforward and effective. However for high aspect surfaces, care must be taken to design a filtering process that removes the effects of noise without obscuring the presence and location of important surface features. The effect of probe

waviness on measured data is a convolution of the probe and the actual surface. Although implemented iteratively as a filtering process, the extraction of corrected surface data from measured data is equivalent to spatial deconvolution. As such, the techniques developed for scanned probe microscopy are relevant to our implementation.

Conclusion

Dealing with probe errors on a polar profilometer can be a challenging endeavor. The unique geometry of the system can, though, become an advantage in the process of evaluating these errors. Once found, misalignment errors and waviness can be applied post-process to correct raw measurement data.

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Web-based Precision Dimensional Verification System for Rapid Design and Manufacture

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Abstract: This paper describes an web-based precision dimensional verification system for rapid design and manufacture. Collaborators related to the development of a new product can confirm geometrical form from the STEP and the inspection data. They can also check dimensional errors, human factors, form errors, as well as mark up the important parts and make a statement of their views over the Internet. Developed system directly uses an inspection file format composed of 3 dimensional point data without any modification of the file format. And for CAD files, STEP is used as a neutral file format. In order to share information between users, this system store dimensional verification and markup results using XML. The usefulness of the developed system is confirmed through a case study.

Key Words: ActiveX, Dimensional Error, Form Error, Human Factor, STEP, Web-based Verification, XML

1. Introduction

In the 21st century, the concept of remote design and manufacture is strongly required in manufacturing processes to reduce cost and time-to-market.

Previously, all the collaborating designers were at the same geographical location within an enterprise. With the evolution of electronic verification tools and wide spread availability of the Internet as a medium for sharing and distributing CAD models and inspection data, the constraint that collaborators should be located geographical proximity has been relieved. The most important activity during the collaborative verification is the resolution of design conflict in the early design stage. This reduces the product development lead-time and manufacturing cost to a large extent.

In the research, Ahn, et al [1] have proposed Internet-based CAD/CAM system. Kan, et al [2] have discuss real time collaborative system for product design in Internet environment using VRML form and Java applet. Huang, et al [3] have begun work in creating a standard Internet based Design for X (DFX) shell that provides a framework in which many types of DFX tools can operate. Commercialized viewing tool that can generate information of product through ST-WebPublisher was developed by Step-Tool Company [4]. However, each of them has insufficient dimensional verification function. They do not support inspection of design and fabrication data.

The objective of this paper is the development of an Internet-based high-precision dimensional verification system for rapid design and manufacture. An inspection client registers inspection data at the developed web server. Collaborators related to the development of a

new product confirm the geometrical forms of inspection data, verify dimensional information and mark up the important parts, and leave messages on their views through the Internet. Functions of the dimensional verification module are constructed as ActiveX by using the visual C++ and OpenGL. This system can check inspection data as well as CAD data to inspect dimensional errors, form errors, human factors over the Internet environment. As we use the inspection data and the STEP file directly, robust verification results are obtained. Dimensional errors of a clay model for a prototype car door are demonstrated to verify the potential effectiveness of the developed dimensional verification system in the distributed business environment.

2. System Architecture

Fig. 1 shows architecture of the verification system. ActiveX-Server accommodates multiple users at the same time. In the architecture, the application server contains dimensional verification module with different functionalities while the client software has the ability to access the server via the Internet [5]. Integration server consists of the database server, the STEP translator and the application server. ActiveX takes charge visualization of inspection data and STEP files as well as dimensional verification results and markup functions. User authentication in the dimensional verification system, file upload and search of the inspection results and STEP files are embodied using Microsoft company's ASP (Active Server Page) and SQL (Structured Query Language).

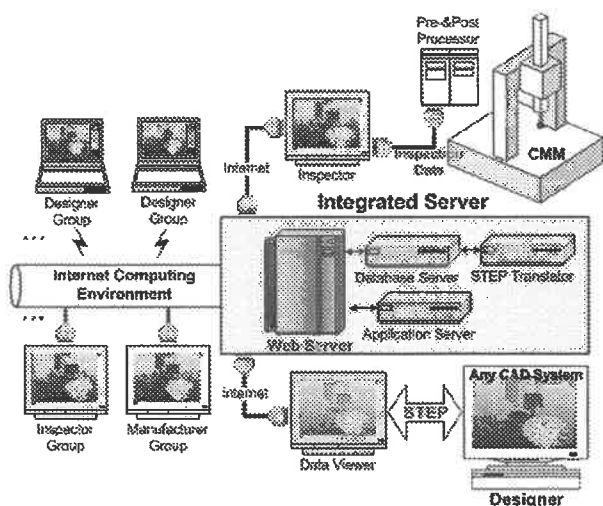


Fig. 1 Architecture of web-based precision dimensional verification system.

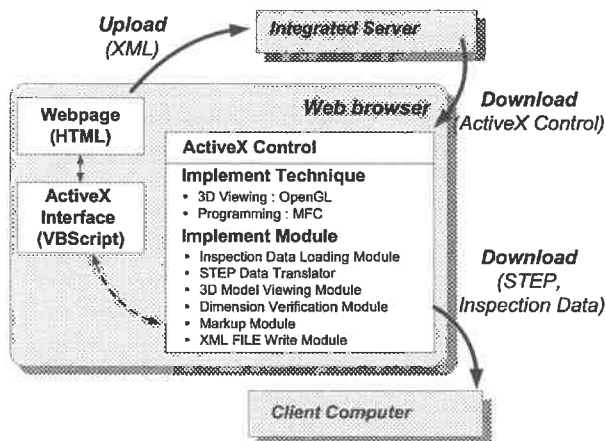


Fig. 2 Configuration of client ActiveX for product verification system.

Developed client ActiveX of the verification system consists of five modules: (1) The inspection data loading module which loads inspection data files. (2) The CAD data translation module extracts features from the CAD/CAM system through the neutral data format of STEP. (3) The 3 dimensional viewing module using 3 dimensional viewing techniques developed by graphics library of Silicon Graphics. (4) The dimensional verification module classifies verification functions according to CAD objects by characterizing geometric entities for high precision dimensional verification. This module has an extensible architecture, so that measurement functions can be easily added to the system. (5) The XML file saving module contains markup and saving functions. Fig. 2 shows the configuration of the client ActiveX.

3. Creation Method of Inspection Data

Most measuring machines have their own data formats. However, a format of inspection data applied to this study is designed to recognize all kinds of formats including not only a format composed of geometric forms and 3 dimensional coordinate values, but also a format just composed of 3 dimensional coordinate information. In general, operators of measuring machines in the styling room know the shape of objects when they digitize clay models of a car. In order to sort digitizing data, they put tick marks between 3 dimensional coordinate values during digitizing processes. They also put tick marks to distinguish geometric features during the digitizing process. By using the above digitizing procedures designers' intention is included in the digitized data. If we inspect the measurement values through clicks of the displayed feature on a monitor, we can confirm the geometric features and verify the dimensions based on the digitized data. In order to be available to verify the dimensions, we put tick marks intentionally when the digitized 3 dimensional values have no tick marks.

4. Geometric Data Extraction through STEP

STEP Standard includes the product data covering the entire product life cycle and has a neutral format that is independent from any software packages and particular hardware platforms. The basic concept to extract geometric features and dimensional information depends upon the STEP physical file containing data instances according to the model schema.

For this purpose, a hierarchical approach is proposed to extract STEP data. Based on this structure, the extraction procedure analyses the STEP file from the highest level of CLOSE_SHELL to the lower level of CARTESIAN_POINT. To ensure the proper connectivity of different functional elements, all the associated pointers are traced in the same manner. Once the necessary data has been extracted from the STEP file, a set of equations representing the feature boundary is obtained. These equations are called boundary equation in [6].

5. Design Method of Dimensional Verification and Markup Module

Coordinate measurement function verifies 3 dimensional coordinate values of a point on the two-dimensional display by selecting a simple mouse click. Measurement of a line length means extraction of the length of line composed of points. The length between two points is measured by the distance difference of two points. Distance between lines is computed from the minimum distance between two lines obtained from

least square fitting of point data. Radius of a circle is computed from a least square fitting of points making a circle. An angle and a curvature composed of 3 points are calculated from the 3-point angle and the radius of curvature. A distance between two circles is computed from the difference of two circle centers.

Mark up function makes sharing information possible between distributed verifiers. Using the mark up function someone on the Internet uploads his or her review messages to the displayed verification feature. Anyone who wants to review others' message about the verification views the mark ups by simple mouse clicks. Fig. 3 shows several mark up functions. Through the mark up functions design and manufacturing collaborations between designers, manufacturers and buyers are available in the distributed environment. They could not only share their opinions about the design but also verify dimension and geometry of the product. Fig. 3 shows available mark up functions. Using the mark up function someone defines his or her viewport and viewing vector on the reviews as well. In order to diversify the sharing of verification results XML file type is incorporated for the developed system. Separators are applied to distinguish dimensional verification results and mark up messages. New dimensional verification result or mark up message is included in the mark up file followed by a separator. In order to allow a lot of consultation from the clients, hyperlink function is also used in the mark up function. It is easy to expand the mark up file by using the separators and hyperlinks [8].

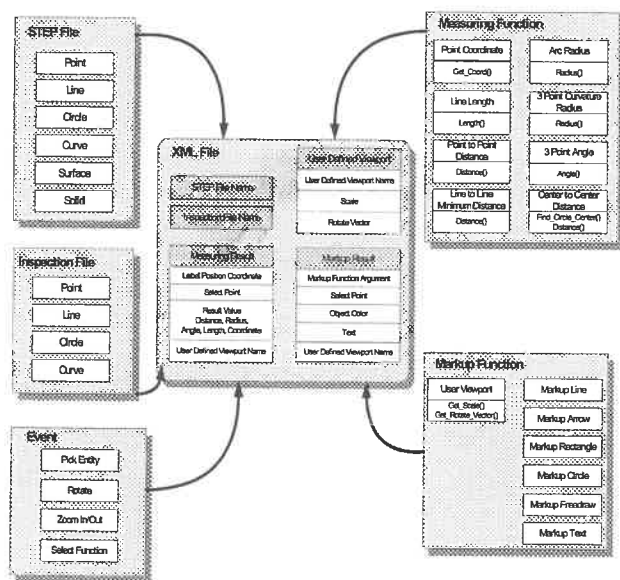


Fig. 3 Architecture of web-based precision dimensional verification system.

6. Case Study

Fig. 4 shows a design and dimensional verification result. After measuring a clay model of a car door with a coordinate measuring machine and inputting a design data through the STEP translator, we can upload the results over the Internet as shown in Fig.4. Clients verify the intermediate design and fabrication result for a reverse engineering over the Internet.

After receiving authentication through the web page, clients such as designers, dimensional inspectors and buyers can get the permission to use the system. In order to start the system, they should input their names, affiliations and date on the starting window. If they select the search file mode, they can look at other clients review files. If they select a specific file, they can review the verification results through the execution of ActiveX.

Fig. 4 shows the dimensional verification according to the design result. From the STEP data the distance between points [A] and [B] of the car door panel is given by 449.002mm as [C]. If we select start point [A] and end point [B] after selecting icon to compute length between points, dimensional verification data of the door panel is given by length 449.150 mm as [D]. By comparing results [C] and [D], user can confirm the design and manufacturing errors. If the client does not satisfy the inspection result, he or she might leave verification message by using the mark up function.

Radius of curvature of a car door is an important human factor. By using the curvature measurement function, user can verify the design and manufacture results. After selecting curvature radius measurement icon, if user clicks points [E], [F] and [G] successively, radius of curvatures are obtained and calculated as 343.006mm of the STEP design result and 344.920mm of the fabrication result as shown in Fig. 4. User can verify the radius of curvatures of the design and fabricated ones.

Fig. 5 shows a design verification example. In this case, user defined viewport is managed by a viewport name specified by the tree structure as [A]. If user selects a viewport name in the treeview, a particular viewport with a specific viewing direction and zoom in/out window is appeared. User can look into the dimensional verification result and markup content as shown in [B] of Fig. 5. Circle and arrow existing in the mark up function as [C] deliver another verifier's message about the current design. Since this mark up function can be added on the verification file as a hyperlink function, clients deliver their opinion in detail and attach related documents and pictures as well.

After reviewing the mark up files written in XML

files, the supervisor of the design and manufacture can modify the current design files or improve the current manufacturing processes through the reverse engineering procedure. He can also upload the improved new design and fabrication results on the Internet for the further reviews.

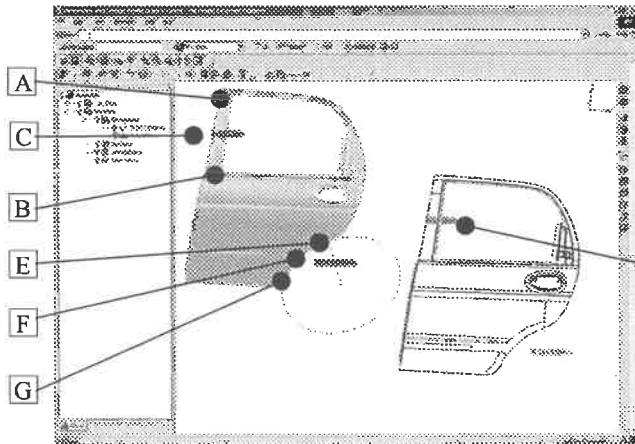


Fig. 4 Case study of STEP and inspection data.

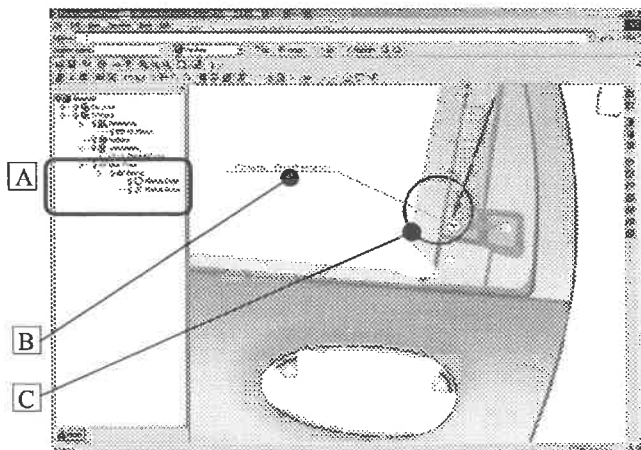


Fig. 5 Case study of design verification.

7. Conclusions

A web-based precision dimensional verification system for rapid design and manufacture is proposed for the collaborative design and manufacture system.

In this system, inspection data and the STEP file are directly used for the verification without modification. It is possible to eliminate file conversion errors and confirm the robustness of the dimensional verification system. This also provides an effective evaluation mechanism to verify dimensions and geometries of the features displayed on the monitor.

By incorporating the mark up function to the system, dimensional verification results and mark up messages

can be easily transferred between verifiers over the Internet. Using the developed hyperlink function and separators in the mark up function, clients can communicate a lot of documents and messages about the verification results each other. In order to expand the information sharing capability, XML file type is introduced to the mark up files.

As the system type is an open-architecture, additional dimensional and design verification functions could be added to the system. This system is an integrated tool that facilitates not only visualization of STEP and inspection data, but also other activities like verification of dimensions. Without the requirement of expensive CAD/CAM hardware or software, users can enjoy a real-time shaded and animated view of their design and digitized models in the remote and distributed environment.

The usefulness of the developed systems has been confirmed through a car door case study.

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A STUDY ON LASER-CCD SPINDLE ERROR MEASUREMENT SYSTEM

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Abstract

Machining accuracy depends on the static and dynamic relative positioning between the spindle and machine table. It is theoretically and practically important to measure the spindle positioning error relative to the machine table. A novel measuring system, which is referred to as the Laser-CCD Spindle Error Measurement System (LCSEMS), is proposed based on the concept of direct and simultaneous measurement for real-time machine error and compensation in regular manufacturing environments.

Keywords: Error compensation, CNC machine tool, Laser, CCD, Spindle error motion

1. Introduction

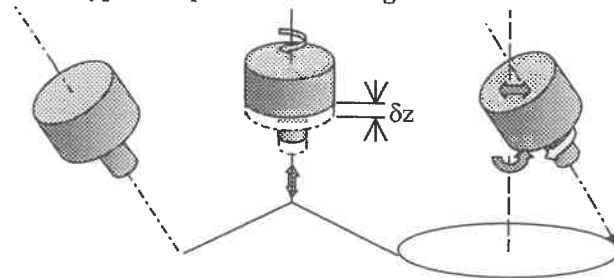
High machining accuracy is one of the most essential factors for modern manufacturing industries. High machining accuracy depends substantially on the reduction of machine tool system errors. According to their sources, machine tool errors mainly include geometric errors, thermal errors, cutting force induced errors and dynamical errors. Many methods and strategies have been employed to reduce these errors are many. According to Blaedel [1], they can be divided into two major categories: error avoidance [e.g. 1, 2-4] and error compensation [e.g. 5, 6].

The basic concept of the error avoidance method is to build accuracy into machine tools and to try to eliminate possible error sources through design and manufacturing efforts. However, this method is limited by several problems such as cost and physical limitation of machine tools. Error compensation is to first measure and then model or map the existing errors of a machine, and finally take compensatory actions during the subsequent machining process to eliminate or reduce these errors. Error compensation method provides possibility to manufacture parts with high accuracy using the machine tools of low accuracy. However, the effectiveness of error compensation practice is limited by the lack of real-time machine error information. Apparently, error compensation would be more effective if real-time machine error information can be obtained. For this purpose, a real-time and direct error measuring system for machine tools is needed.

A machine spindle rotates at high speed to cut workpieces and it slides along the rotational axis for positioning. It is very important to be able to detect the erroneous motion of the spindle because it is the part that comes in contact with the workpieces. The goal of this research is to develop a system, called Laser-CCD Spindle Error Measurement System (LCSEMS) that can measure several major motion errors of one machine

spindle simultaneously, in-process, and in real-time. In this paper, the major erroneous features are identified. The required accuracy and test methods by ISO are mentioned. The theoretical basis for simultaneously measuring the errors has been established. A system prototype has been designed and a simulation to find the optimal parameters for the design has been performed.

2. Types of Spindle Positioning Errors



Inclination Thermal Expansion Radial Error Motion

Figure 1 Three types of spindle errors studied

There are four types of spindle positioning errors: inclination, thermal expansion, four-degree-of-freedom radial motion error and four-degree-of-freedom linear motion error.

Inclination is where the spindle is tilted at an angle to its ideal orientation while at rest, therefore a static error. A spindle will expand when its temperature rises. There are many causes that might raise the temperature of a spindle: change in room temperature, electrical current, or friction. Some of the causes occur when the spindle is at rest and others occur when it is in motion; therefore, thermal errors can either be static or dynamic. The other two errors are dynamic errors. Four-degree-of-freedom radial motion error occurs when the spindle is in rotation. Ideally a spindle should be rotating against a single point; however, it is in reality rotating in a circular path around the ideal point of rotation due

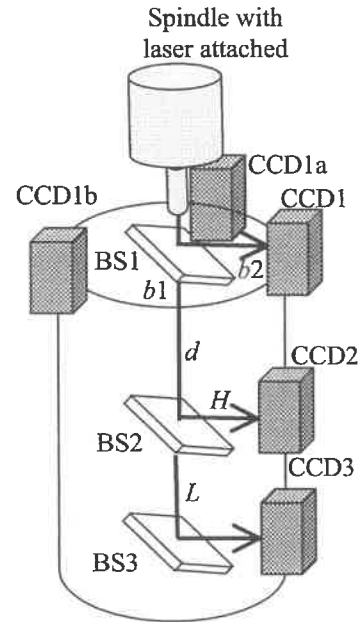
to errors introduced by spindle. Four-degree-of-freedom linear motion error occurs when the spindle is sliding along the rotational axis linearly. Instead of moving in a perfectly straight line, there are some unwanted four degree-of-freedom motions incurred in the motion. Figure 1 shows the three types of errors this research is concerned. Linear motion error measurement was covered by previously research in IMS lab [7], though the current system should be able to measure it as well.

3. Conceptual Design of the System

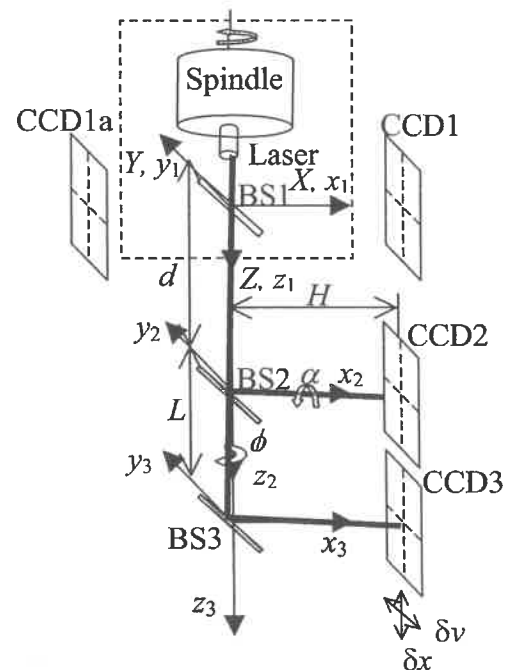
LCSEMS has two main modules: a laser module and a CCD module. Laser module includes a laser and a beam splitter. The laser is attached to the tip of a machine spindle. One beam splitter is fixed at the tip of the laser. CCD module includes five CCD cameras and two beam splitters. The laser provides a stable laser beam, which can be detected by the CCD cameras. When the spindle is in motion, either rotating or sliding, the laser moves with the spindle. The laser beam image captured by the CCD cameras will reflect the movements of the laser or spindle. Knowing the correlation between the laser beam movement and its image on the the CCD cameras, we can determine the exact movement of the spindle. To ensure the simplicity and efficiency, the system should be designed in such a way that it will be able to measure all errors simultaneously and in real time. Also, the system should be portable for different types of CNC machines.

Figure 2(a) shows the conceptual design of the system. LCSEMS has two main modules: a laser module and a CCD measuring module and a computer system. The laser module consists of a single laser and a beam splitter (BS1) to provide dual laser beams as the two-metrology bases for LCSEMS. One laser beam (b1) lies in Z-axis, and the second laser beam (b2) lies perpendicular to the Z-axis. The CCD module consists of five CCD cameras: CCD1, CCD1a, CCD1b, CCD2, and CCD3; two beam splitters, BS2 and BS3, are also included in the CCD module. The CCD module will be rigidly fixed on the machine table in the reference frame. As shown in Figure 2(b), uppercase X, Y, Z is the coordinate system of the spindle and laser module; lowercase x, y, z is the CCD module coordinate system, which is fixed in the reference frame.

H is the shortest distance between the CCD cameras and z-axis, L is the distance between BS2 and BS3 or CCD2 and CCD3, and d is the distance between BS1 and the CCD module. The beam positions on each CCD will change when the spindle rotates or moves up and down along the z-axis. The spindle inclination, thermal expansion, straightness errors δx and δy , and angular errors α and ϕ can be determined simultaneously from the beam positions on each CCD.



(a) Schematic of the system



(b) Coordinate systems of LCSEMS

Figure 2 Conceptual Design of LCSEMS

4. Measurement Principles

In Figure 2(b), $X-Y-Z$ is the moving coordinate system attached at the center of BS1. There are three global coordinate systems: $x_1-y_1-z_1$, $x_2-y_2-z_2$ and $x_3-y_3-z_3$. $x_2-y_2-z_2$ and $x_3-y_3-z_3$ are attached to the center of BS2 and BS3 respectively $x_1-y_1-z_1$ is attached to the ideal center of BS1. In Figure 2(b), $X-Y-Z$ and $x_1-y_1-z_1$ coincide each other.

4.1 Inclination

The coordinate system X - Y - Z moves with respect to the global coordinate system x_1 - y_1 - z_1 . Because BS1 is fixed on and rotates with the spindle, laser beam b2 is always perpendicular to the axis of the laser. The three centers of CCD1, CCD1a and CCD1b form a triangle on the plane of x_1 and y_1 . Since we are measuring the orientation of the spindle at rest, we can manually rotate the spindle so that laser beam b2 is pointed towards each of the three CCDs and collect the beam position on each CCD. The three positions of laser beam image on CCDs form a plane, in which we have the coordinate information needed to determine the spatial relation of the spindle with respect to the global reference frame x_1 - y_1 - z_1 (as shown in Figure 3).

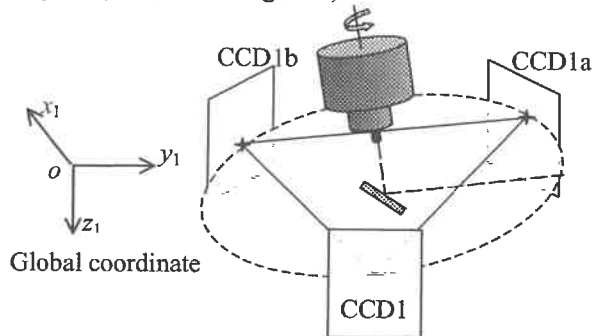


Figure 3 Measurement principle of inclination
Spindle positions pre/post thermal expansion

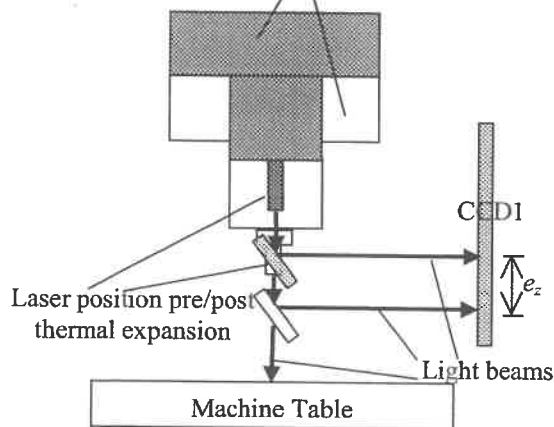


Figure 4 Measurement principle of thermal expansion

4.2 Thermal expansion

In order to detect the thermal expansion effect on a spindle, we need to measure the spindle tip position over time. The position of laser beam b2 is first measured on CCD1 while the spindle is at rest and room temperature. To inspect the thermal expansion effect on a spindle in a given time and a given RPM, the spindle is then turned on for the given time period at the given speed. A second beam position is then taken. The displacement of beam position between the two

beam positions is the result of thermal expansion for the given time and speed as shown in Figure 4).

4.3 Four-degree-of freedom radial motion error

Radial error motion has six components. Three translational components: δx , δy and δz , and three rotational components: α , ϕ , and θ . The six components result in a six degree of freedom erroneous motion when rotating the spindle. From the past research [7], δz cannot be measured with good resolution. Therefore, it will not be considered in this research. Also, θ does not affect workpiece accuracy due to the symmetrical geometry of spindles. Therefore, it will not be considered, either. As a result, only four-degree-of-freedom motion errors will be measured in this research.

Based on the conceptual system design, the spatial correlation between the spindle tip and the positions of the laser beam image on CCD2 and CCD3 was derived by vector analysis as follows

$$\begin{bmatrix} y_2 \\ z_2 \\ y_3 \\ z_3 \end{bmatrix} = \begin{bmatrix} 0 & -1 & H & 0 \\ -1 & 0 & 0 & -H \\ 0 & -1 & H+L & 0 \\ -1 & 0 & 0 & -(H+L) \end{bmatrix} \begin{bmatrix} \delta x \\ \delta y \\ \alpha \\ \phi \end{bmatrix} \quad (1)$$

From the inversion (Equation (2)) of this equation, we can obtain the four erroneous components.

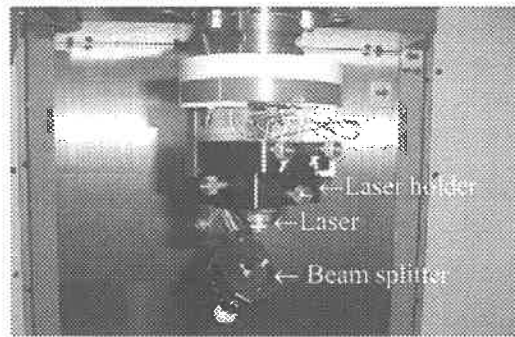
$$\begin{bmatrix} \delta x \\ \delta y \\ \alpha \\ \phi \end{bmatrix} = \begin{bmatrix} -\frac{H+L}{L}z_2 + \frac{H}{L}z_3 \\ -\frac{H+L}{L}y_2 + \frac{H}{L}y_3 \\ -\frac{1}{L}y_2 + \frac{1}{L}y_3 \\ \frac{1}{L}z_2 - \frac{1}{L}z_3 \end{bmatrix} \quad (2)$$

When the spindle rotates, the coordinate system X - Y - Z moves with respect to the global coordinate systems x_2 - y_2 - z_2 and x_3 - y_3 - z_3 . The relative motion can be decomposed into two parts: a two-dimensional translation of δx and δy and a composite rotation α and ϕ . Once beam center position (y_2, z_2) and (y_3, z_3) on the two CCDs are captured, the four-degree-of-freedom machine motion error components (δx , δy , α , ϕ) can be found from Equation (2). The coordinate (y_2, z_2) and (y_3, z_3) are the beam positions on CCD2 and CCD3 respectively. L and H are the structural parameters defined in Figure 2.

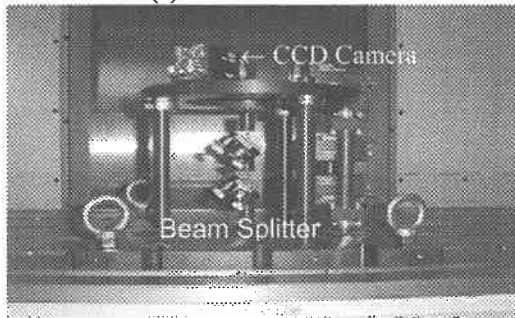
5. Prototype Evaluation of LCSEMS

Based on the above analysis, a prototype of this system has been built. As aforementioned, the system includes

two module. Figures 5(a) and (b) show the pictures for these two modules.



(a) Laser module



(b) CCD module

Figure 5 Picture of system prototype

5.1 Laser module

One of the major requirements for the laser module is to have stable beam generation. This is because the accuracy of the system depends on the laser beam to provide an accurate reference point. Therefore, checking the laser beam stability becomes an important priority for system evaluation. Environmental vibration, temperature change and even stability of laser power supply will influence the laser stability. Steps taken to test laser beam stability include beam profile detection, beam drift calculation and beam processing. To numerically increase laser beam stability, a process called "frame-averaging" is used. This process adds a certain number of beam images together before dividing the image by the number to take the average beam intensity distribution. Then the beam center is calculated using the Centroid method. For final measurement, average of a certain number of beam center point will be used to increase the accuracy of measurement. Figure 6 shows the result of averaging every 100 points out of 1500 beam center points. The variation for the 15 resulted center points is about 0.3 μm , which meets the 1 μm accuracy requirement.

5.2 CCD module

CCD camera has noise known as "dark current". To minimize the effect of this noise, dark current's highest level is measured and then subtracted from the data collected.

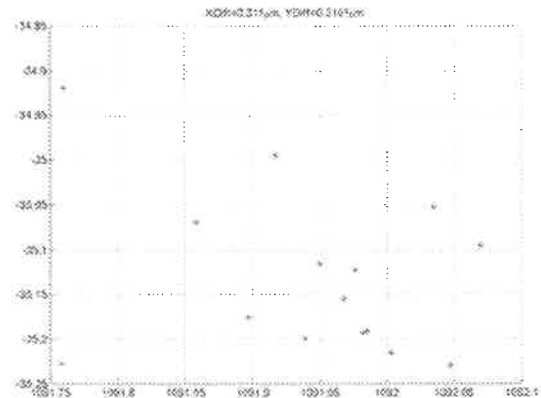


Figure 6 Averaged center point

6. Conclusions and Future Work

A system called Laser-CCD Spindle Error Measurement System (LCSEMS) has been proposed to measure four types of spindle positioning errors. The measurement principles have been discussed in detail. A prototype has been built and major problems associated with measurement accuracy have been identified. Initial evaluation indicates that the system can meet the 1 μm accuracy requirement.

Future work is to improve the stability of laser and therefore to maintain the measurement accuracy of the system, and finalize the prototype.

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Performance Evaluation of a Prototype Machine Tool for Machining Meso-scaled Parts

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Abstract

With the increased prevalence of meso-scaled products and feature sizes, new tools are needed to bridge the gap between fabrication processes tailored for micrometer sized features and those for millimeter sized features [1-4]. Compared to its traditional counterpart, a small machine tool designed for meso-scale parts has the advantages of: ease of use, smaller footprint, smaller structural loop, shorter distances between the workpiece and the machine's metrology devices, opportunities for improved machine metrology, and easier environmental control. At the same time, the small work volume presents a challenge when measuring the machine's geometric errors, as traditional machine tool metrology instruments (i.e. lasers, ballbars, levels, etc.) are difficult if not impossible to use in very small work volumes. This paper describes a prototype three-axis meso-scale machine tool that has been built and is configured for rotary milling applications. In addition, preliminary performance evaluation results for this machine will be presented. The project's is focused on developing new metrology tools and methods for meso-scale machine tools with small work volumes. This prototype machine is one of three meso-scale machine tools under development within the Machine Tool Metrology Group at the National Institute of Standards and Technology (NIST).

Machine Design and Configuration

The machine tool implementation described in this paper is that of a prototype three axis meso-scale mill configured for rotary milling applications (e.g. production of camera zoom lens bodies). This machine is a partial realization of a five axis machine design [5]. The prototype machine is designed for a 25 mm cubed work volume. The workpiece is supported by a THK* ballscrew-spline that is driven by a DC brushless motor coupled by a cable-capstan drive transmission. This configuration provides 60 mm of linear motion and 340 degrees of rotary motion of the workpiece. In addition, the prototype machine includes a high speed spindle which has 10 mm of linear motion along the spindle axis of rotation. A voice coil actuator with a linear encoder is used to close the servo loop that positions the end mill tool mounted in the spindle. The spindle is orthogonal to the ballscrew-spline shaft (and its linear motion) as shown in Figure 1 (a) and (b).

The prototype machine is unique in the use of a ballscrew-spline to support and position a workpiece in place of slideways used on traditional machine tools. For the meso-mill application, the ballscrew-spline shaft resembles a large lead ballscrew shaft with additional linear ball grooves ground along the length of the shaft. Located on the shaft are two ball nuts (one ballscrew nut and one ball-spline nut) whose outside flanges are bolted to opposite ends of a precision ground rectangular cast iron fixture. As the ball screw nut is rotated, with the ball spline nut fixed, the shaft translates linearly. As the ball-spline nut is rotated, with the ballscrew nut fixed, the shaft both rotates and translates. By coordinating the rotary motions of both nuts, it is possible to obtain only rotary motion or only pure linear motion of the shaft. This coordination is achieved through the use of the coordinate system capabilities on the motion control board. This enables the user to program the machine using traditional linear and rotary program statements.

The ball nut is fixed within the mounting flange through ball bearings captured in two opposing angular contact ball bearing raceways. The outside diameter of the ball nut has opposing angular contact bearing raceways machined on itself and the inside diameter of the mounting flange has corresponding bearing raceways machined in its internal diameter. In order to drive the nuts, a 305 mm machined drum is bolted to each of the ball nut end faces. Two

* Commercial equipment, instruments, or materials are identified in this report in order to specify adequately certain procedures. In no case does such identification imply recommendation or endorsement by the National Institute of Standards and Technology, nor does it imply that the material or equipment identified is necessarily the best available for the purpose.

brushless servo motors, with integral resolvers for motor shaft position feedback, are used to drive the two ball nuts using a cable transmission mechanism. A 3.2 mm diameter multi-strand steel cable wraps around a 51 mm diameter pulley on the motor shaft. The cable crosses over itself and wraps around the drum described above. From an end view, the cable wrap looks like a figure eight and a tensioning mechanism internal to the drum provides the necessary preload. This figure-eight configuration increases the rigidity of the cable drive mechanism [5].

Metrology Issues

The small work volume of this machine presents a challenge when measuring geometric errors. Traditional machine tool metrology instruments (i.e. lasers, ballbars, levels etc.) are difficult, if not impossible, to use in such a small work volume. A focus of this research effort at the NIST is to develop new metrology tools and methods to characterize small work volume meso/micro scale machine tools.

The novel application of a ballscrew-spline transmission mechanism as a slideway also presents a challenge. The motions of the ballscrew-spline shaft (and the workpiece) are constrained only by the recirculating balls of the ball nuts and the angular contact bearings between the ball nut and mounting flange as previously described. From a metrology standpoint, the ballscrew-spline shaft acts as a rotary and linear machining axis, and it becomes important to measure the changes in the orientation and position of the axis of rotation. Currently, the workpiece position is indirectly determined by the resolvers integral to the servo motor shafts. We are evaluating mounting a tape scale to the inside of the drum for possible use in a more direct measure of the workpiece. In addition, the tape scale could be used to evaluate the linearity of the cable transmission design.

Measurement Results

The performance characterization of the machine will include the assessment of the geometric errors associated with the linear motion of the spindle and the rotary and linear motion of the ballscrew-spline transmission mechanism (at the workpiece). As well as errors in the critical alignment between the axes. The results from measuring the horizontal straightness, the vertical straightness, and the pitch and yaw errors of the ballscrew-spline motion are shown in Figures 2 – 5. The measurements were performed using two capacitance probes offset by 25 mm in a holder attached at the location of the workpiece. The machine was programmed to move 60 mm forward, then reverse 60 mm, while the capacitance probes measured deviations with respect to a straight edge. The straight edge artifact error is determined by the reversal method, and the artifact error is subtracted from the measured results. Using the two offset probes, one can measure an angular error while simultaneously measuring the straightness error. The pitch error is measured with the vertical straightness and the yaw error is measured with the horizontal straightness.

The output of the analog capacitance probe is digitized the motion control board. By using this interface, the motor shaft position and the capacitance probe outputs are simultaneously sampled. This eliminates the problems of synchronizing data collection on multiple instruments with the motor shaft position. From Figure 2, the vertical straightness error is 0.30 μm having a measurement uncertainty of 0.03 μm using a coverage factor $k=2$. From Figure 3, the horizontal straightness error is 0.54 μm having a measurement uncertainty of 0.03 μm using a coverage factor $k=2$. It is possible that the recirculating balls in the ball nuts are causing the periodic error motions, as a ball in a helical track must pass over an axial track and vice versa. From Figure 4 the range of the yaw error is 0.62 arc-seconds having a measurement uncertainty of 0.2 arc-seconds using a coverage factor $k=2$. From Figure 5 the range of the pitch error is 1.8 arc-seconds having a measurement uncertainty of 0.2 arc-seconds using a coverage factor $k=2$. From these preliminary results, the prototype machine has the potential to meet the accuracy goal of $\pm 0.5 \mu\text{m}$. Hence despite the balls having to cross tracks, the ballscrew-spline system has potential as a combined actuator/bearing/axis structure for meso-scale machines.

Summary

A prototype three axis-rotary milling machine tool for machining meso-scale parts has been built and is currently undergoing testing. The unique machine configuration presents a number of metrology challenges. Preliminary tests show on a ballscrew-spline that this machine element has potential for use as a precision actuator/bearing/axis structure for meso-scale machines. Ballscrew-splines have been used in robotic pick-and-place mechanisms for many years, but not in short-stroke high-precision applications as a slideway substitute on a machine tool.

Acknowledgments

The authors would like to acknowledge the contribution of Wojciech Kosmowski and Mark Kosmowski of ESI for donating the machine's spindle, and Kevin Perkins of THK for donating the ballscrew-spline.

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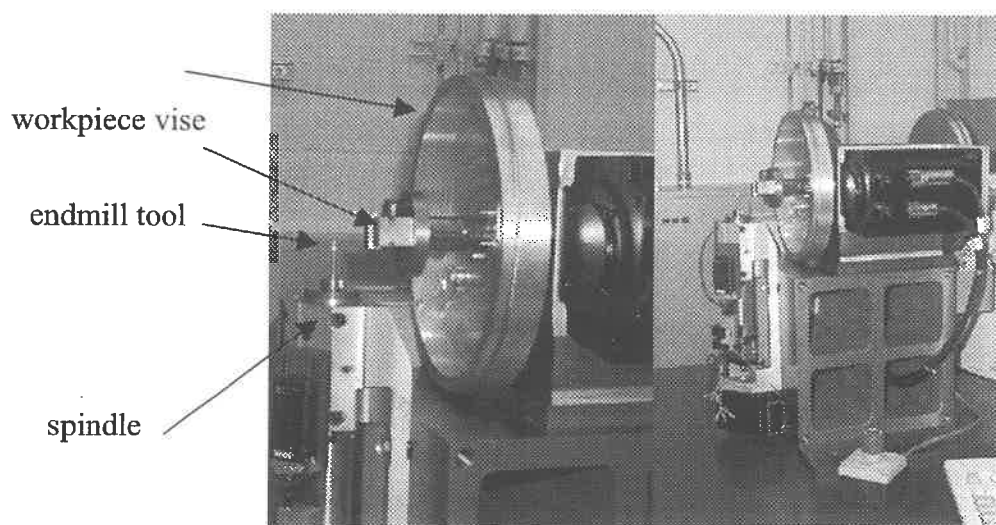


Figure 1 (a) and (b), Photos of the prototype three axis rotary milling machine tool (enclosure removed)

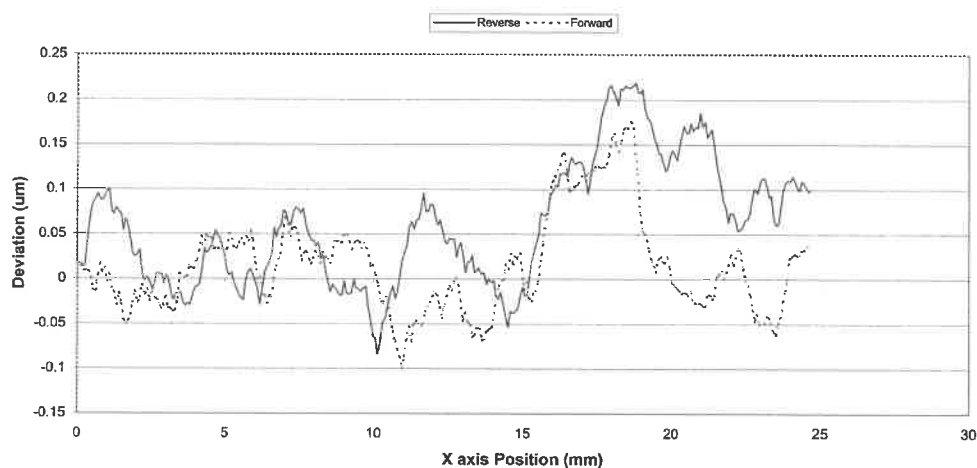


Figure 2, Plot of the prototype machine's vertical straightness error measured at the workpiece location.

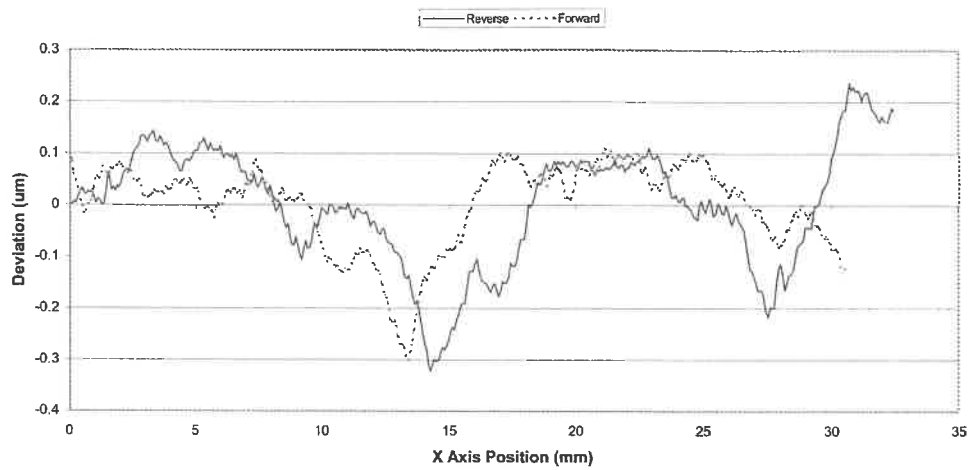


Figure 3, Plot of the prototype machine's horizontal straightness error measured at the workpiece location.

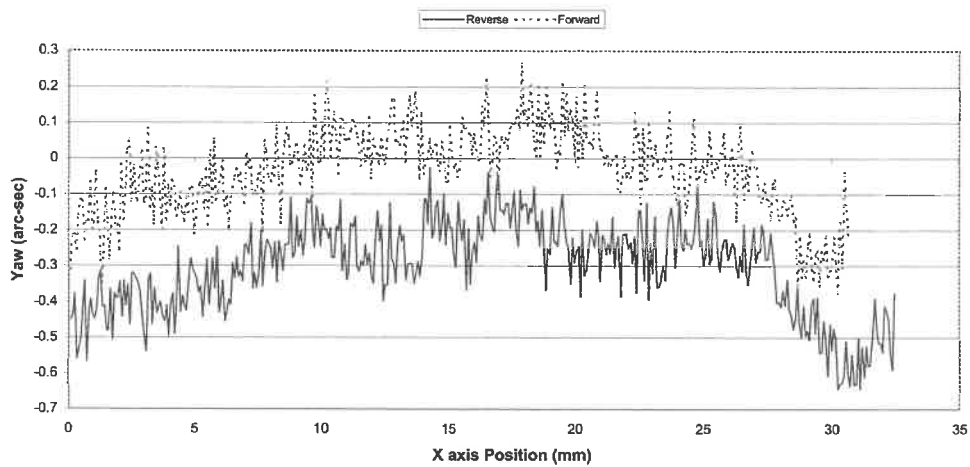


Figure 4, Plot of the prototype machine's yaw error measured at the workpiece location.

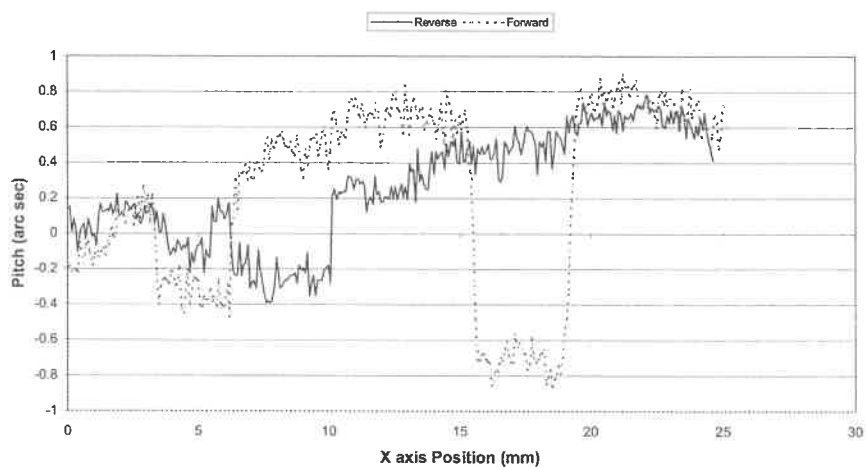


Figure 5, Plot of the prototype machine's pitch error measured at the workpiece location.

Measurement of Systematic Geometric Errors in Coordinated Measuring Machines.

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The measurements carried out in the industry require, for their complexity and quantity, the use of Coordinated Measuring Machines (CMMs). This type of machines possesses inherent geometry errors due to its construction, the dominant causes of these errors are: the imperfections that presents the guides that moves the axes X, Y and Z of the CMM, the displacements of their transducers, the probe system and the adjustment to each other of the components of the machine. Additionally, these errors are influenced by the temperature variations.

For the case of Cartesian CMMs, twenty one errors of geometric nature plus the errors of the probe system, should be taken into account to improve the accuracy of the same ones. The big CMMs (>2000 mm) have additionally, the so called "errors of non rigid body".

The method that is presented was developed to measure and to correct the errors of CMMs of big dimensions of "rigid body" and of "non rigid body", for this are used unidimensional elements (ball beam, spherical beam or cylindrical beam). This method includes the measurement procedure that should be employed and a software that carries out the calculation of the geometrical errors.

Introduction

There are several methods that have been used for the geometrical errors measurement of CMMs, between them the following ones can be enumerated: method of the ball plate[1] and classic methods using laser, electronic levels and reference patterns, however none embraces the CMMs of big dimensions completely. The method of the ball beam was developed to cover big CMMs as well as small ones. This new method is based partially on the same ideas that are used in the ball plate method, but it contains another form for the calculation of the roll errors [2], since with this method certain problems are avoided that are observed in the ball plate method.

Description of the ball beam method (21 errors for CMM of "rigid body").

The method uses basically trigonometrical and geometrical principles, beside some classic methods already exist to calculate the 21 errors.

Firstly all the rotation errors : yaw and pitch, are evaluated by the error difference divided by the distances between the positions of the beams, and the lengths of the stylus, see figure 1.

Yaw (XRZ): Rotation of the Z axis that affects the position of the X axis.

$$XRZ = \frac{X2(X) - X1(X)}{a}; a = X2(Y) - X1(Y) \quad (1)$$

Pitch: (XRY): Rotation of the Y axis that affects the position of the X axis.

$$XRY = -1 \cdot \left(\frac{X3(X) - X1(X)}{b} \right); b = X3(Z) - X1(Z) \quad (2)$$

Yaw (YRZ): Rotation of the Z axis that affects the position of the Y axis.

$$YRZ = -1 \cdot \left(\frac{(Y3(Y) - Y4(Y)) + (Y1(Y) - Y2(Y))}{2a} \right); a = Y3(P) - Y4(P) \quad (3)$$

Pitch: (YRX): Rotation of the X axis that affects the position of the Y axis.

$$YRX = \frac{(Y3(Y) + Y4(Y)) - (Y1(Y) + Y2(Y))}{2b}; b = Y3(Z) - Y1(Z) \quad (4)$$

Yaw (ZRY): Rotation of the Y axis that affects the position of the Z axis.

$$ZRY = \frac{Z1(Z) - Z2(Z)}{b}; b = Z1(P) - Z2(P) \quad (5)$$

Pitch (ZRX): Rotation of the X axis that affects the position of the Z axis.

$$ZRX = \frac{Z3(Z) - \left(\frac{Z1(Z) + Z2(Z)}{2} \right)}{a}; a = Z3(P) \quad (6)$$

Where: X, Y and Z are the Cartesian coordinates of each one of the balls of the beam and P it is the length of the stylus in a certain position. Example: Y3(Y) represents the value of the coordinate Y in the position Y3, Z3(P) represents the length of the stylus in the position Z3, they are not functions but positions.

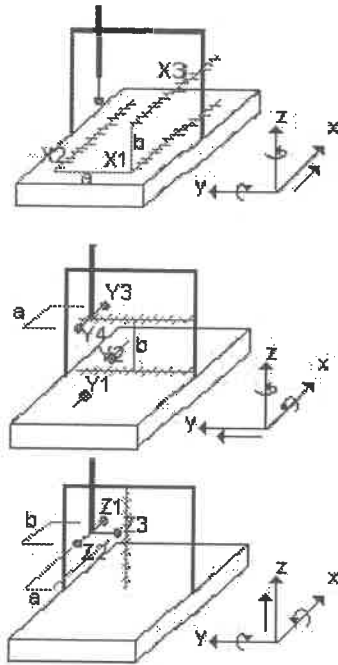


Figure 1. Positions of the ball beam to evaluate the yaw, pitch and position errors.

The position errors are directly evaluated from the length errors in the parallel positions to the axes and then referred to the axes of the coordinate system using the corresponding errors of rotation, with the distance of an axis of the coordinated system to the line of measurements of the beam.

Position (XTX): translation of the X axis that affects the position of the X axis.

$$XTX = X_{cal} - \{X1(X) - (XRZ \times X1(Y)) - (XRY \times X1(Z))\} \quad (7)$$

Position (YTY): translation of the Y axis that affects the position of the Y axis.

$$YTY = Y_{cal} - \left\{ \frac{(Y1(Y) + YRX \times Y1(Z)) + (Y3(Y) + YRX \times Y3(Z))}{4} + \frac{(Y2(Y) + YRX \times Y2(Z)) + (Y4(Y) + YRX \times Y4(Z))}{4} \right\} + \left\{ \frac{-YRZ \times Y1(P) - YRZ \times Y2(P) - YRZ \times Y3(P) - YRZ \times Y4(P)}{4} \right\} \quad (8)$$

Position (ZTZ): translation of the Z axis that affects the position of the Z axis.

$$ZTZ = Z_{cal} - \left\{ \frac{(Z1(Z) + ZRY \times Z1(P)) + (Z2(Z) + ZRY \times Z2(P))}{2} \right\} \quad (9)$$

The roll errors in X and Y are evaluated from the differences of straightness derived from measurements carried out in parallel positions of the beam respect to the axes of the CMM, dividing the result between the distance of the beams. This calculation is carried out only after correcting the orientation of the ball beam nominally parallel. This correction is carried out with the measurements of the ball beam in diagonal position. These diagonals (three balls measured in each diagonal position) contains the flatness deviation information of the geometry of the CMM in the four balls of the ends that form the diagonals [2]. The two parallel beams to the axes that are located in the respective plane of the CMM (X1 and X2, X1 and X3, Y1, Y2 and Y3, Y4), are aligned mathematically, in such a way that the end ball of each beam, follow the flatness error obtained from the diagonals, see figure 2 and 3. Note that the end ball of each beam should coincide in an approximate way with the positions of the end balls from the diagonals.

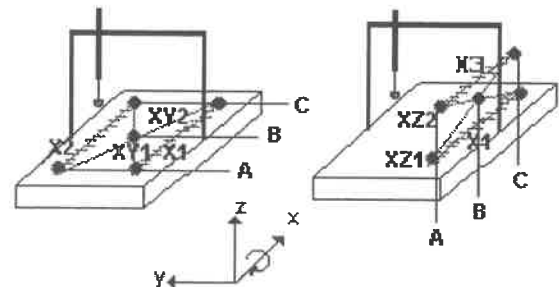


Figure 2. Positions of the ball beam to evaluate the roll errors and straightness of the X axis.

Correction of the coordinate Z of the balls measured in the position X2.

$$Z_{corr} = -1 \cdot \left\{ \frac{(XY1C(Z) - XY2C(Z)) - (XY2A(Z) - XY1A(Z)) - 2 \cdot (XY1B(Z) - XY2B(Z))}{2} \right\} \quad (10)$$

$$\left\{ \frac{(X2C(Z) - X1C(Z)) - (X2A(Z) - X1A(Z))}{2} \right\}$$

Correction of the coordinate Y of the balls measured in the position X3.

$$Y_{corr} = -1 \cdot \left\{ \frac{(XZ1C(Y) - XZ2C(Y)) - (XZ2A(Y) - XZ1A(Y))}{2 \cdot (XZ1B(Y) - XZ2B(Y))} \right\} - \left\{ \frac{(X3C(Y) - X1C(Y)) - (X3A(Y) - X1A(Y))}{2} \right\} \quad (11)$$

Roll (XRX): Rotation of the X axis that affects to the same axis.

$$XRX = -1 \cdot \left\{ \frac{X2(Z_{corr}) - X1(Z) + X1(Y) - X3(Y_{corr})}{2} \right\} \quad (12)$$

Correction of the coordinated X of the balls measured in the positions Y3 and Y4.

$$X_{corr} = -1 \cdot \left\{ \frac{(YZ1C(X) - YZ2C(X)) - (YZ2A(X) - YZ1A(X))}{2 \cdot (YZ1B(X) - YZ2B(X))} \right\} - \left\{ \frac{(Y_{3,4}C(X) - Y_{1,2}C(X)) - (Y_{3,4}A(X) - Y_{1,2}A(X))}{2} \right\} \quad (13)$$

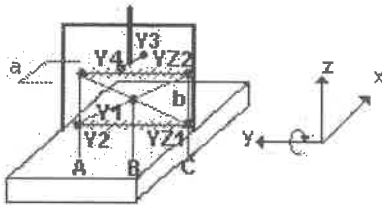


Figure 3. Positions of the ball beam to evaluate the roll errors and straightness of the Y axis.

Roll (YRY): Rotation of the Y axis that affects to the same axis.

$$YRY = -1 \cdot \left\{ \frac{(Y3(X_{corr}) + Y4(X_{corr})) - (Y1(X) + Y2(X))}{2b} \right\} \quad (14)$$

The roll error in the Z axis of the CMM, is calculated from the difference of straightness when the beam is measured in a parallel position to the Z axis, using two different lengths of stylus. See figure 1.

Correction of the coordinate Y, for the positions Z1 and Z2.

$$Z1(Y) = Z1(Y) - \{XRX(Z1(X) - Z1(P)) \times Z1(Z)\} \quad (15)$$

$$Z2(Y) = Z2(Y) - \{XRX(Z2(X) - Z2(P)) \times Z2(Z)\} \quad (16)$$

Roll (ZRX): Rotation of the Z axis that affects to the same axis.

$$ZRX = \frac{Z2(Y) - Z1(Y)}{Z1(P) - Z2(P)} \quad (17)$$

The errors of straightness are directly evaluated from the parallel beams to the axes, correcting the errors of straightness, because of this, the errors are valid in the coordinate axes (the correction is the roll error multiplied with the distance from the beam to the axis of the respective coordinate).

Straightness (XTZ): Straightness of the X axis in the direction of the Z axis.

$$XTZ = Z_{cal} - \{X1(Z) + (XRX(X) \times X1(Y_0))\} \quad (18)$$

Straightness (XTY): Straightness of the X axis in the direction of the Y axis.

$$XTY = Y_{cal} - \{X1(Y) - (XRX(X) \times X1(Z_0))\} \quad (19)$$

Straightness (YTX): Straightness of the Y axis in the direction of the X axis.

$$YTX = X_{cal} - \left\{ \frac{Y1(X) + Y2(X)}{2} + (YRY(Y) \times Y1(Z_0)) \right\} \quad (20)$$

Straightness (YTZ): Straightness of the Y axis in the direction of the Z axis.

$$YTZ = Z_{cal} - \left\{ \frac{Y1(Z) + Y2(Z)}{2} + (YRY(Y) \times Y1(P)) + (YRY(Y) \times Y2(P)) \right\} \quad (21)$$

Straightness (ZTX): Straightness of the Z axis in the direction of the X axis.

$$ZTX = X_{cal} - \left\{ \frac{Z1(X) + Z2(X)}{2} \right\} \quad (22)$$

Straightness (ZTY): Straightness of the Z axis in the direction of the Y axis.

$$ZTY = Y_{cal} - \left\{ \frac{(Z1(Y) - ZRX(Z) \times Z1(P)) + (Z2(Y) - ZRX(Z) \times Z2(P)) + Z3(Y)}{3} \right\} \quad (23)$$

Finally the perpendicularity errors are evaluated from the difference of lengths of the diagonals measurements on each plane of the coordinate system (see figures 2 and 3). However, these lengths are only evaluated after correcting the coordinates of the respective balls with all the calculated errors, applying in a correct way the stylus vectors used during the measurement of the diagonals.

Perpendicularity (XWY): perpendicularity between the X axis and the Y axis.

$$XWY = \frac{(LXY2_{corr} - LXY1_{corr}) \times LXY1_{corr}}{2 \times U(XY1) \times V(XY1)} \quad (24)$$

Perpendicularity (XWZ): perpendicularity between the X axis and the Z axis.

$$XWZ = \frac{(LXZ2_{corr} - LXZ1_{corr}) \times LXZ1_{corr}}{2 \times U(XZ1) \times V(XZ1)} \quad (25)$$

Perpendicularity (YWZ): perpendicularity between the Y axis and the Z axis.

$$YWZ = \frac{(LYZ1_{corr} - LYZ2_{corr}) \times LYZ1_{corr}}{2 \times U(YZ1) \times V(YZ1)} \quad (26)$$

Where: L represents the length of the measured diagonals with the end balls (length of the beam), U and V are the lengths of each diagonal projected to each axes. Example: LXY2corr is the length of the diagonal XY2 corrected by all those errors except those of perpendicularity and V(XZ1) is the length of the diagonal XZ1, projected on the Z axis.

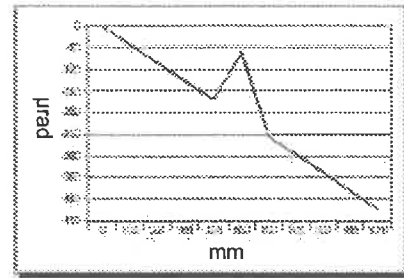
CALMAC, version 1.0.

This program which initials make reference to CMM calibration with Beams was developed to calculate the 21 errors of CMM of "rigid body" and the three errors of CMM of "elastic body". This version includes measurement temperature correction, robust interpolation, union of beams for the case when the beam length doesn't cover the total volume of the CMM and the obtention of the correction format developed by the PTB of Germany.

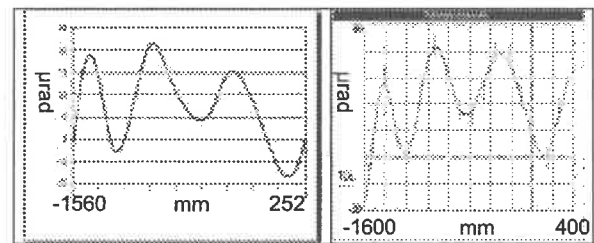
Tests of validation for CALMAC.

Two tests were carried out: one of them was based on synthetic data, studying the capacity to analyze each error by separate, and a test in which the results of a real measurements with the ball beam method were compared with the measurements carried out with the

method of the ball plate [1] (making use of the program KALKOM 4.0) with the same CMM.



Figures 5. Roll Error in the Y axis with a lineal content and a "local pulse". The results went identical to the results that in theory should be obtained.



Figures 6: Comparison of results between the ball plate method, plate of 960 mm x 960 mm (right) and a ball beam of 3 m (left). The evaluation was carried out with KALKOM 4.0 from the PTB and with CALMAC 1.0. Here the roll errors of the X axis are shown.

Conclusions.

This method is of great utility not only to evaluate and to correct CMM geometrical errors with the objects method, but also has the flexibility of being able to be used to evaluate this errors making use of the classic methods that use laser interferometer, or the relatively new methods as the laser tracker to evaluate and to correct errors. Especially the program Calmac 1.0, can be adapted to the use of these instruments.

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Data Mining of Conditions for Machine Tools Based on Machining Information

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1. Introduction

With the increasing enhancement of product accuracy and reducing machining costs in the recent years, demands for functions which can predict machining conditions in advance for CAM systems, and controllers of machine tools (otherwise called CNC (Computerized Numerical Control) systems) are growing stronger and stronger. In actual machining, it is difficult to predict machining load from the performance of workpieces, tools, machine tools, etc. Highly precise measuring instruments for monitoring machining conditions and dedicated controllers are therefore mounted on the machine tools. However, such equipment is expensive and one equipment cannot be mounted onto several machine tools. Since machining conditions can be grasped using NC data generated by CAM (Computer Aided Manufacturing) in simulations for predicting machining load, in this research, we built a simulation system which can monitor the states of machine tools easily and inexpensively.

2. Outline of system

Fig. 1 shows the configuration of the system. It is composed of machine tools, machine tool controller (CNC), and PC (Personal Computer). To analyze machining conditions, the CNC is connected to the PC by high-speed serial bus (HSSB). A machining condition analysis program, CAM, machining load simulator, and machining database are equipped in the PC. The machining condition analysis program performs acquisition and analysis of machine tool state data, and optimization of NC data. The acquired CNC data and analysis data are stored in the machining database. The machining load simulator selects the cutting feed rate, and detects tool breakage using NC data generated by CAM.

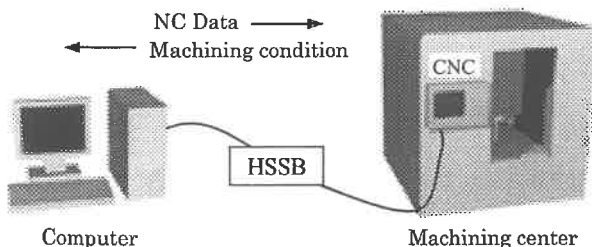


Fig.1 System configuration

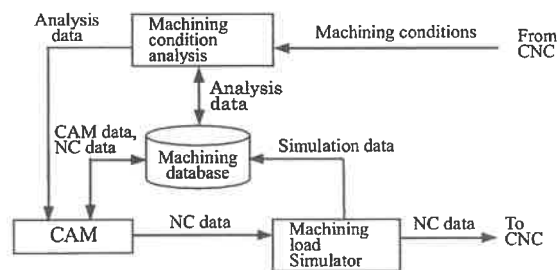


Fig.2 System configuration of PC

3. Cutting conditions for machining condition analysis

Machining condition data in plane cutting was acquired using a ball end mill with a diameter of 10mm. The depth of cut was set at 0.25, 0.5, 1.0, 1.5, and 2.0mm. The feed rate was changed to 2000, 3000, 4000, 5000, 6000, and 7000 mm/min according to these depths. To acquire CNC data in convex cutting, a ball end mill with a diameter of 10mm was used in the same way. The angle of inclination was set at 45, 60, and 75 degrees, and the depth was cut was varied to 0.25, 0.5, and 1.0mm according to the angle of inclination. A vertical high-speed milling machining center mounting a FANUC 160iM onto a controller was used in all the cutting jobs. Machining tool state data was acquired at intervals of 10msec. The acquired data consists of the machining load of the principal axis, the number of rotations, servo adjustment current, and machine coordinates of the XYZ axes.

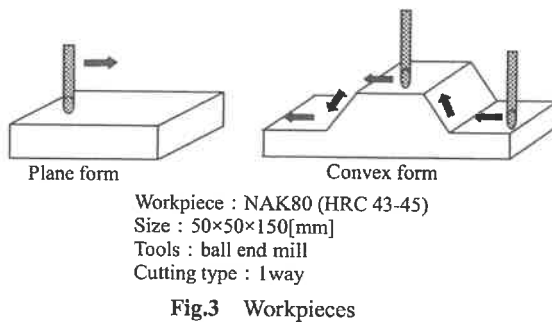


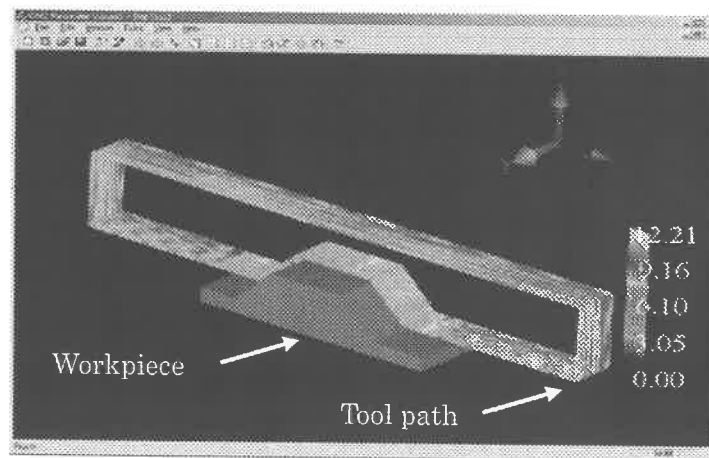
Table 1 Cutting conditions

Cutting conditions		
Plane form	Feed rate	2000-7000[mm/min]
	Spindle speed	3000-12000[rpm]
	Axial depth of cut	0.25-2.0[mm]
Convex form	Feed rate	3000[mm/min]
	Spindle speed	10000[rpm]
	Axial depth of cut	0.25-1.0[mm]

4. Experimental results

The machining condition analysis program was created to display the tool path and machining load together to clearly indicate different values and changes in respect to complicated tool paths.

As shown in Fig. 4, the operating screen displays the tool path according to the changes in the color displayed from the machine tool state data, allowing changes in the data to be checked visually. Moreover, cutting forms can be read, and operations like changing the viewpoint and zooming can be performed using the mouse. The analysis data is stored in the machining database with the cutting form, and can be read easily. Fig. 5 shows an example of analysis of the machining load during cutting.



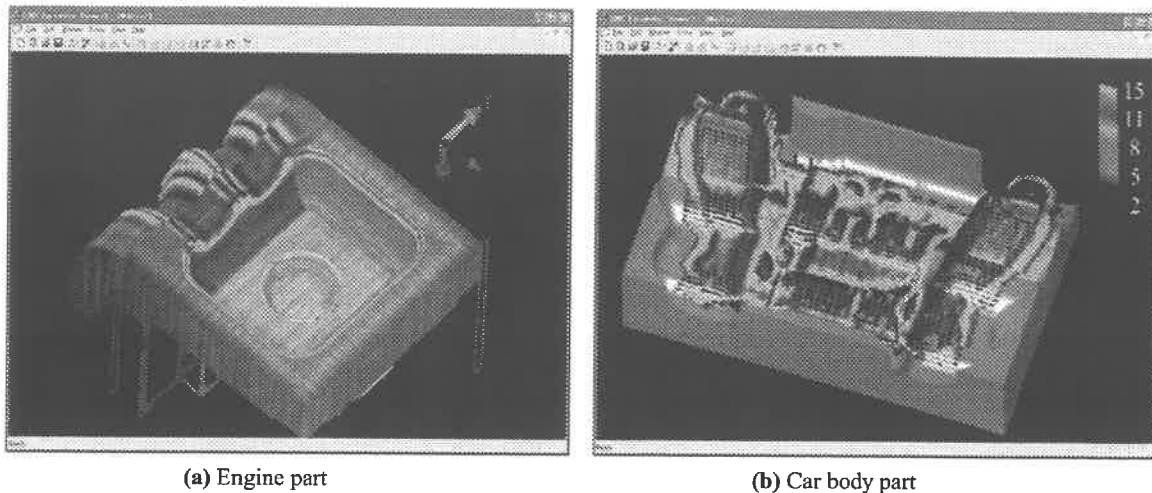


Fig.5 Example of machining load analysis

Fig. 6 shows the comparison results of prediction load and real load in volume cutting calculated by the machining load prediction simulator. Load prediction was performed by changing the cutting volume every second while cutting, and regression-analyzing the machining load at this time. Consequently, the difference between the prediction load and real load was about 4% maximum. Cutting volume per second is used for this prediction to use the cutting volume output from the NC machining simulator. More accurate prediction should be possible in the future by the mining of machining data according to various cutting conditions. Fig. 7 shows the comparison results of prediction load and real load in convex machining. Prediction is carried out based on cutting volume per second on the NC simulator. Comparison of the prediction load and real load showed good agreement as an overall tendency. However, two major peaks were seen for prediction load but not for real load. Some of the possible reasons for this may be due to the CNC data acquisition cycle, operating speed of CNC, etc., since the position where the load peak appears can be confirmed clearly, tool breakage can be prevented, and machining accuracy improved.

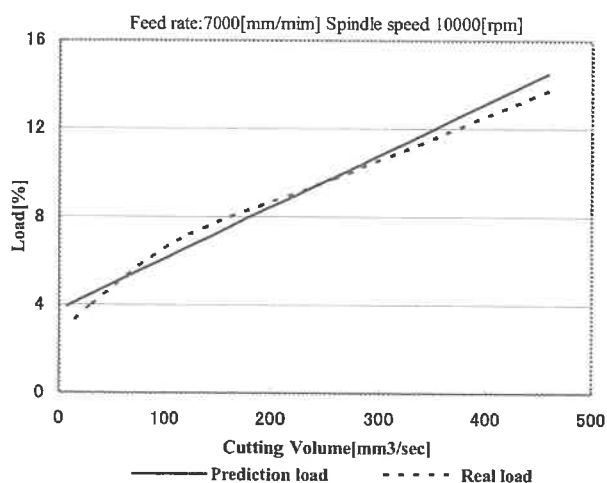


Fig.6 Comparison results of prediction load and real load

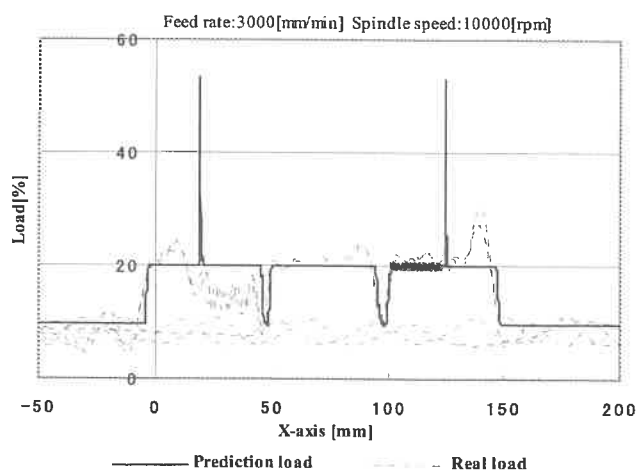


Fig.7 Comparison of prediction load and real load in convex machining

5. Conclusions

In this study, a system which can predict machining load by acquiring machining condition data during cutting from CNC was built and the following conclusions were obtained.

- (1) Machining conditions during cutting can be obtained from CNC at regular intervals using the machining condition analysis program.
- (2) By linking the tool path and machine tool state data, machining load can be checked visually. Moreover, positions where load is large could be detected from the relation between the tool position and machining load.
- (3) Machining load can be predicted by carrying out machining under various conditions, storing machine tool state data in the machining database, and analyzing the data.

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INSPECTION STRATEGY FOR LIGA MICROSTRUCTURES USING A PROGRAMMABLE OPTICAL MICROSCOPE

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INTRODUCTION

The LIGA process uses deep X-ray lithography and electroforming to create high aspect ratio microstructures. This technology is well established at Sandia National Laboratories (SNL) [1,2]. Dimensional metrology for these microparts is crucial for process control and to assure dimensional control. However, the metrology of LIGA parts remains a challenging task, with much room for development [3]. This study is intended to address the uncertainty issues that high aspect ratio mesoscale structures with microscale features present in non-contact dimensional metrology using a programmable optical microscope. Inspection quality of this type is dependent on what the camera senses, as well as what the edge detection algorithms are able to decipher. As the manufacturer specifies submicron resolution and accuracy [4], this precision may be difficult to achieve due to the high aspect ratio specimens at hand. Calibration standards for this type of machine are composed of thin chrome on glass, having virtually no thickness ($\sim 0.1\mu\text{m}$ anti-reflective chromium) and no topology in comparison to LIGA structures. Objects of this type are used to properly assess the attainable precision of the optics as well as the stage accuracy of this tool. One the other hand, structures made of materials native to the LIGA lithography process, such as negative photoresists, positive photoresists, and electrodeposited metals, require distinctive microscope settings for proper inspection.

Instructions for setting lighting conditions typically have guided the user to use low levels of light. Using light sparingly avoids flooding the sensor so that image saturation or blooming does not occur. A study on how edge detection is effected by a change in lighting type and intensity is performed on chrome and high aspect ratio specimens.

According to research performed at Lawrence Livermore National Labs regarding the strengths and weaknesses of metrology tools, the optical microscope suits the inspection scale desired for LIGA metrology [5]. The microscope used in this study is a VIEW Engineering Voyager V6x12 [5]. The hardware and the software of this tool are commercially available, and are used as supplied by the vendor. Reiterating, the purpose of this study is to find out how to most accurately measure high aspect ratio structures using a programmable optical microscope.

EXPERIMENTAL SETUP

In LIGA microfabrication, we typically measure the linewidth of a channel or line. Therefore, a NIST and NPL certified chrome linewidth standard ($\pm 0.266\mu\text{m}$) was chosen to be the first specimen to undergo testing.

The test consists of selecting a range of lighting intensities for each lighting type offered by the microscope. The ranges are chosen so that an image is produced that has enough contrast for the edge detection algorithm to execute. Measurements were taken so that both boundaries of the channel were within the field of view of the camera during inspection so that stage uncertainty was removed. Ten measurements were taken at pitches, distances marked by alternating chrome-to-glass transitions, of $50\mu\text{m}$, $100\mu\text{m}$, $150\mu\text{m}$, and $200\mu\text{m}$ for lighting intensities from 19 percent to maximum intensity, in 5 percent increments. Since coaxial (top) lighting was unable to produce acceptable contrast, only back lighting and red LED ring lighting from the overhead were used on the chrome standard. Figure 1 shows the results of these measurements for the $50\mu\text{m}$ pitch.

Deviations of Voyager from 50um Linewidth Standard (MRS3) with respect to Lighting Intensities and Type at 20x Magnification

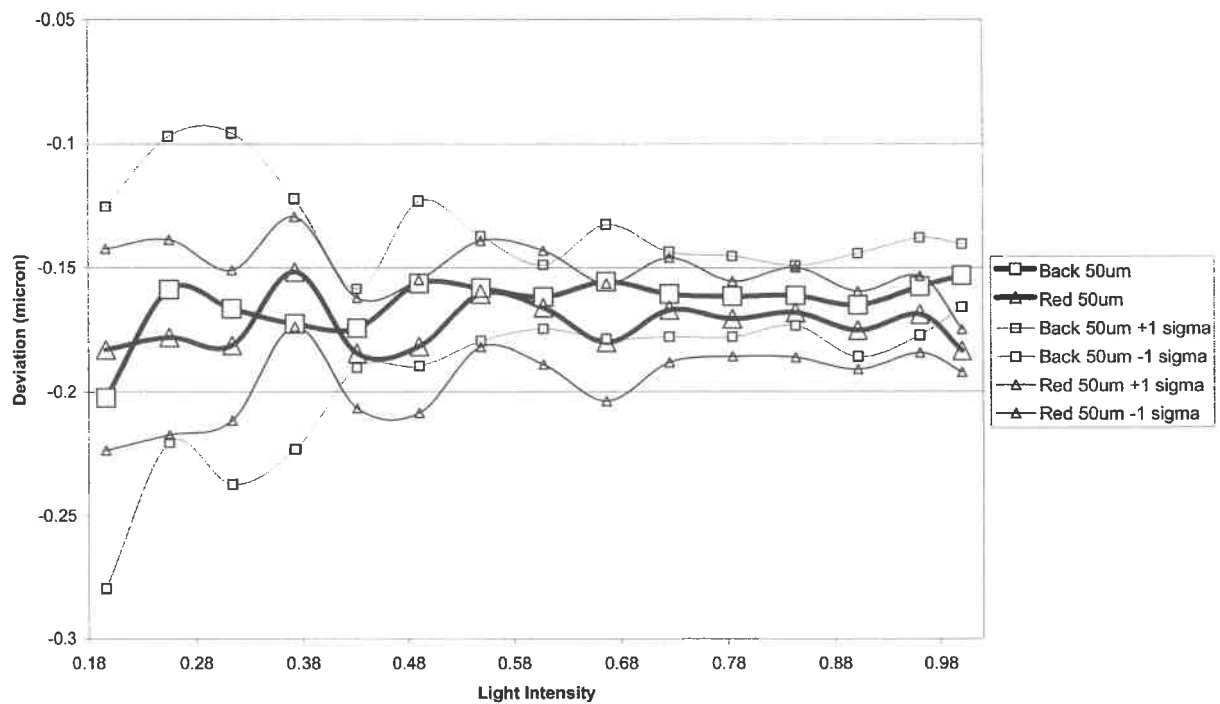


Figure 1: Lighting Study results from linewidth standard

As shown in Figure 1, the thicker lines represent the mean values of the ten measurements taken at the specified lighting intensity. The standard deviations (thin lines) show that as the lighting intensity reaches about 50 percent of maximum, the measurements become more consistent. As in all inspection processes, reproducibility is the coveted goal. Figure 1 also shows that noise is reduced for light intensities greater than 50 percent. The results of averaging values are shown in Table 1. The results show little difference in the measurements with respect to the lighting type choice.

Table 1: Linewidth Standard Results (Back Lighting and Red LED Lighting: 50% - 100%)

	Mean (μm)	Standard Deviation (μm)
Back Light	-0.159	0.019
Red LED Light	-0.172	0.019

To most closely emulate a LIGA processed metal structure, the first case study consists of a 150 μm NIST traceable stainless steel gauge block (certified to $\pm 0.04\mu\text{m}$). This specimen has been encased in clear epoxy polymer and then planarized by lapping and polishing procedures (as used in our baseline LIGA process). Embedded in the epoxy, it is subjected to inspection under a range of suitable lighting intensities and lighting types offered by the microscope. Figure 2 shows a screen shot of the gage block using each type of light.

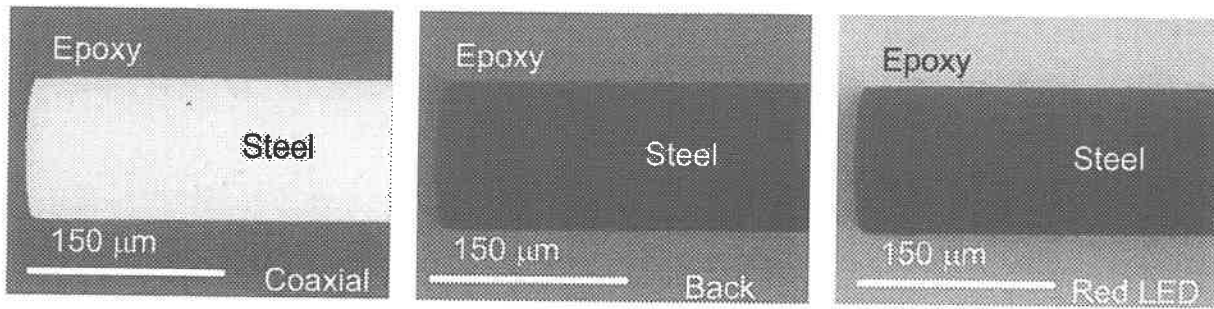


Figure 2: Embedded gage block at 20x magnification under coaxial, back, and red LED lighting conditions (arbitrary intensities)

The same lighting and measurement procedure was performed on this specimen and the results are shown in Figure 3.

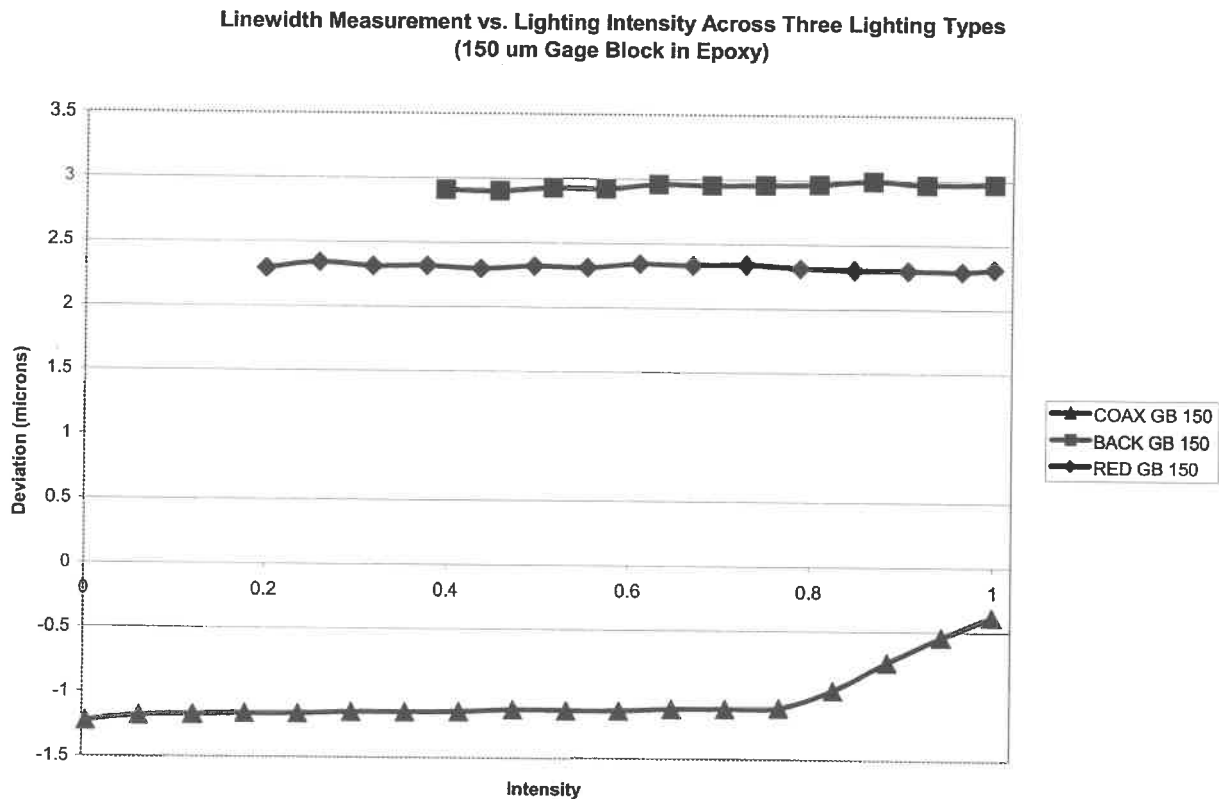


Figure 3: Lighting Intensity Study Performed on 150μm Gage Block

A much larger difference in measurement is seen between the back lighting and the red LED lighting. This variation may be attributed to the high aspect ratio of the structure as well as the edge topology created by the material removed in the lapping and polishing processes. Standard deviations exhibit the same trends across lower lighting levels.

The deviations seen in Figure 3 are beyond the specified precision ($\pm 0.04\mu\text{m}$) of the certified gage block. This deviation is probably due to changes in the gauge block caused by the lapping and polishing. After being released from the epoxy, the same lighting procedure was performed, and SEM measurements calibrated to the linewidth standard were used to verify the gauge block measurements with greater precision. SEM results are

compared to the microscope measurements to ensure that the most accurate lighting environment may be used in future inspections of embedded and released metallic parts.

CASE STUDIES (continued)

The gage block aids in understanding the “released part” step in LIGA. However, in order to aid process improvement with metrology, not only the final step in production must be covered. For example, X-ray mask and polymer mold, occurring earlier in the process line, must also be inspected. Some of the high aspect ratio structured materials are SU-8 and PMMA, negative and positive photoresists. Both materials are transparent polymers bonded to radiant metal films on a polished substrate. This combination of materials, together with a finite radius of the edge results in a dark line on the order of $2\mu\text{m}$ thickness. Figure 4 shows both polymers under inspection.

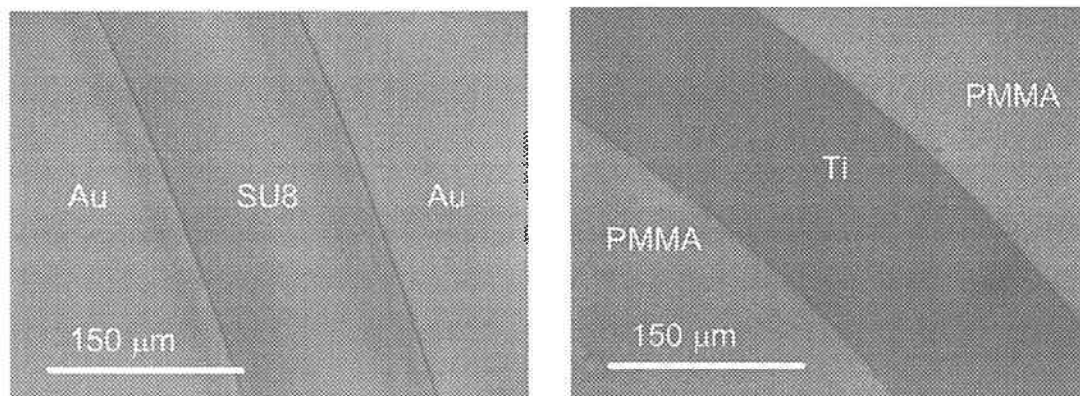


Figure 4: SU8 on Gold (Left) and PMMA on Titanium (Right) both at 20x Mag

A more precise measure of where the “proper” edge lies is once again performed by using SEM strategies on $150\mu\text{m}$ prepared samples. The results are compared to measurements made using various lighting and edge detection conditions with the Voyager.

CONCLUSIONS

By implementing this LIGA inspection strategy, a superior understanding of the precision of the microscope as well as the nuances of inspecting high aspect ratio microstructures is attained. This gives greater confidence in the accuracy of inspection results generated by the programmable optical microscope. This benefits LIGA technology by enabling rapid and precise inspections across all process steps, generating both direct measurements and coordinate point clouds so that further post-processing may be completed and inspection history (raw coordinate data from edge detection) may be electronically archived.

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Measurement Uncertainty in Scanning Instruments Due to Data Acquisition Methods

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Introduction

The objective of this study is to advance the understanding of measurement uncertainty in horizontal scanning measurement instruments. Such instruments are widely used in a variety of industries to measure form, fit, and surface texture on scales ranging from nanometers to meters. They are currently available from several manufacturers, and use sensors and motion control elements having different characteristics to implement their measurements. The measurement modes used in these instruments inherently affect the uncertainty in measurements. As the speed, accuracy, and repeatability of measurements become increasingly critical, identifying, quantifying and minimizing such measurement uncertainty take on increased industrial relevance.

Approach

Research on measurement uncertainty typically focuses on instrument calibration (Garnaes, *et. al.*, 2003; Qian, *et. al.*, 2000; Bienias, *et. al.*, 1997; Mainsah, *et. al.*, 1995) or on instrument comparison (Ohlsson, *et. al.*, 2001). Yan and coworkers have decomposed the sources of geometric error, studied the effect of measurement uncertainty and suggested methods of reducing variation in a three-part paper (Yan, *et. al.*, 1999). Literature searches revealed little information on the effect of software design and instrument configuration on measurement uncertainty. Therefore, this study focuses on the effect of the design of the software used to control the instrument and the design of the data acquisition function.

Equipment

A 3-axis scanning laser microscope (SLM) was used to identify and quantify sources of measurement uncertainty. The SLM was originally built with support from NASA to measure the fine scale texture on runway surfaces and was designed to be easily reconfigurable. It uses three CompuMotor™ stages (Parker-Hannafin, CA) coupled to quadrature encoders and has a horizontal range of 150 x 150 mm and a vertical range of 66 mm. A Keyence™ LC-2210 triangulation height sensor (Keyence Corporation, NJ) is used to measure the topography and has an experimentally determined lateral resolution of 25µm and vertical resolution of 12 µm (Johnson, 1997). A Sensoray 421™ (Sensoray Inc., OR) data acquisition card was used to acquire laser height readings. The stages were configured to move at a near-constant velocity while the triangulation sensor records the height at predetermined locations for this study. The SLM was controlled by a PC running Microsoft™ Windows™ 2000. These hardware and software configurations were chosen as they allowed relatively easy changes in the implementation of data acquisition and instrument control.

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Results and Discussion

Figure 1 shows the effect of software thread priority on the interval between successive data points. The charts represent a single trace at a velocity of 10 mm/s, with a desired measurement interval of 25.4 μm . The chart on the left shows the achieved measurement interval under normal thread priority for the data acquisition loop. Large spikes appear, as other running threads scavenge processor cycles from the data acquisition thread. When the thread priority is changed to real-time, (Figure 1, right), all other threads are preempted by the data acquisition thread. Under these conditions, the data acquisition thread is able to acquire data at consistent intervals.

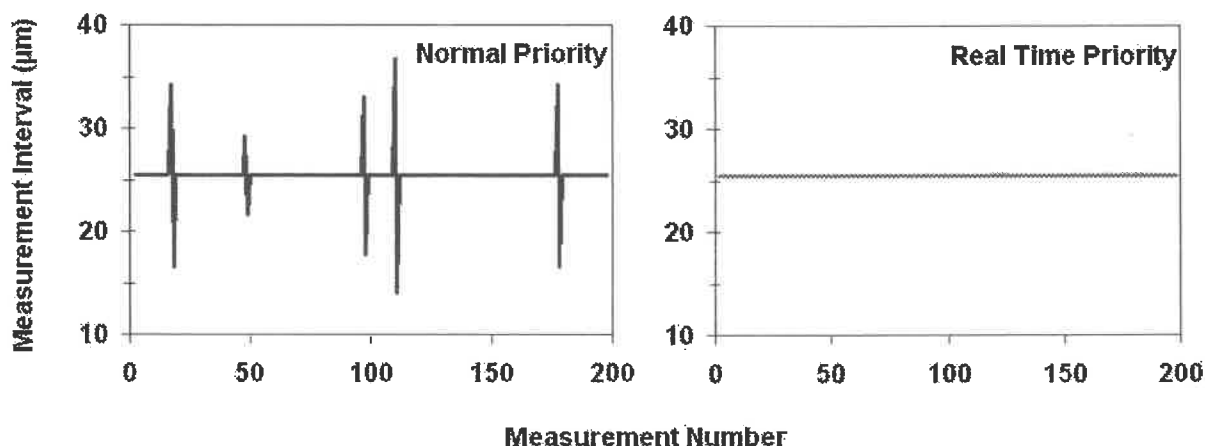


Figure 1: Effect of thread priority of the data acquisition loop.

Left: The operating system may unilaterally preempt the data acquisition thread in normal priority mode, leading to large ($\sim 1800 \mu\text{m}$) intervals between data points. Right: No such preemption was noticed under real-time priority.

There may be conditions under which the number of operations being executed in the data acquisition loop changes. This temporarily affects the data acquisition rate, increasing the uncertainty in the measurement. Figure 2 shows the effect of varying the effective length of the data acquisition loop by requiring the software to dynamically allocate memory. Two memory buffers were compared: one, with a size of 0.2 MB and another with a size of 20 MB. The data being written to the buffer was 16 bytes long, so the smaller buffer was guaranteed to fill up before the measurement ended. In both cases, the desired measurement interval was set at 10.16 μm .

The results show that the average measurement interval is equal to the desired measurement interval of 10.16 μm in both cases. However, in the case of the smaller buffer, (Figure 2, left), the operating system was forced to dynamically allocate memory every time the memory buffer filled up. This periodically increased the number of instructions in the loop, increasing the measurement uncertainty. The measurement uncertainty was seen to decrease as the scan velocity was increased because fewer data points were collected at higher velocities, requiring fewer memory allocations. For the larger buffer, (Figure 2, right), the instruction length of the data acquisition loop effectively remained constant, as the buffer never needed to be replenished. Thus, the instrument was able to consistently meet the required measurement interval and the standard deviation was zero across the velocities tested.

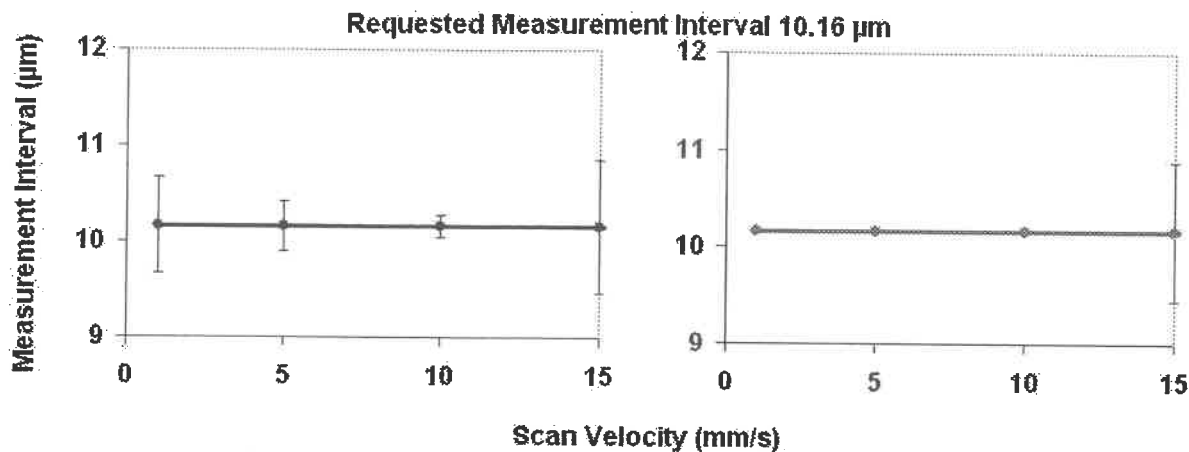


Figure 2: Effect of varying data acquisition loop length

The operating system allocates memory for the acquired data. Left: memory must be allocated frequently at slower speeds. Right: The buffer is pre-allocated, so the data acquisition proceeds unhindered.

A measurement mode often used is to have the vertical axis “follow” the surface. In this mode, the axis moves vertically whenever it crosses a predefined threshold, attempting to keep the surface within the range of the measuring laser. Surface following may be achieved using either software or hardware control. In the former case surface following functional requirement competes with data acquisition functional requirement, while in the latter case, it is independent or decoupled (Suh, 1990; Suh, 2000).

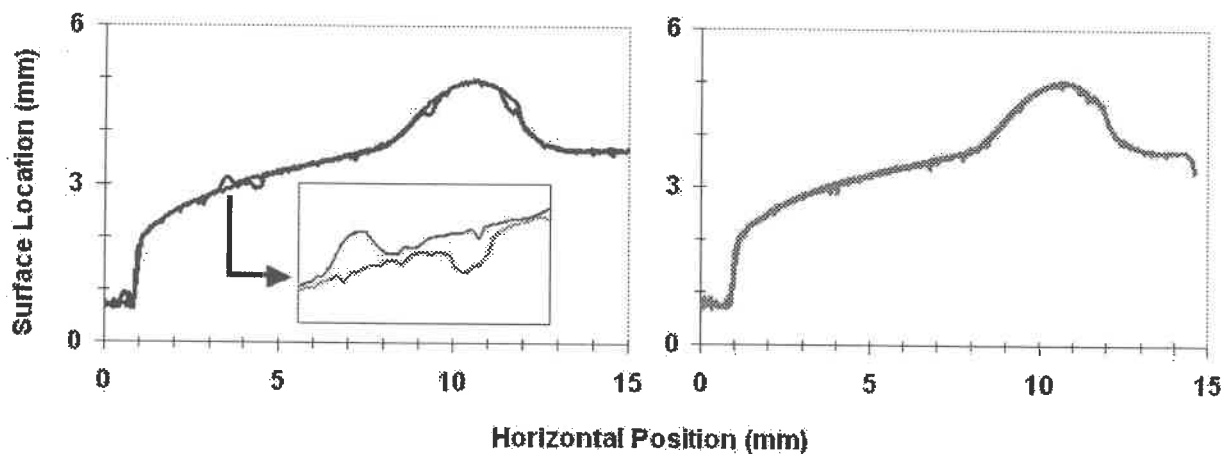


Figure 3: Effect of decoupling functional requirements

The instrument was designed to “follow” the surface being measured. Left: surface following is dependent on data acquisition loop to initiate vertical axis movement, increasing measurement uncertainty. Right: surface following is decoupled from data acquisition loop and implemented by hardware, decreasing measurement uncertainty.

Figure 3 shows the effect of decoupling axes-related functional requirements from data acquisition. Each figure consists of two lines – one for the forward trace and one for the reverse.

When the vertical axis movement is triggered by software, (Figure 3, left), the number of instructions to be processed in the loop temporarily increases every time the threshold is reached. The height reading

reported, therefore, does not exactly correspond to the horizontal location. As a result, a peak appears when the vertical axis is moving away from the surface while a trough appears when the vertical axis is moving towards the surface (Figure 3, inset).

When the vertical axis movement is triggered by hardware, (Figure 3, right), vertical axis motion is triggered using “near” and “far” hardware triggers. The motor controller moves the vertical axis independently upon sensing either of these events. Since this does not affect the data acquisition, which is implemented using software, the measurements mirror surface features well.

Conclusions

This study has demonstrated how the control software design affects the measurement uncertainty of scanning instruments. It has also demonstrated that the data acquisition loop in the code must be decoupled from the other functions of the equipment to reduce the measurement uncertainty.

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UNCERTAINTY OF MEASUREMENTS USING MULTISENSOR CMMs

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INTRODUCTION

Hybrid, or multisensor, CMMs are growing in popularity due to the flexibility of changing the sensing device without having to re-establish the part's coordinate system. This allows us to determine a datum reference frame from physical features that may be measured (for example) with a touch trigger probe, and then inspect small or fragile features with a non-contacting system, while maintaining the datum reference frame already determined. While the assessment of the errors or uncertainty present in the use of each probing system is much like that described in the existing B89.4.1 standard, the quality of measurements relying on points taken with different sensors is more complex.

In this paper we report our implementation of proposed changes to the B89.4.1 standard as an initial assessment of the Hybrid CMM performance. In addition, we discuss testing methods that will expose errors or changes in the relative offsets of the different sensors. The primary focus of our work is a better understanding of the uncertainty introduced when measuring the same work piece with different sensors (dubbed "sensor fusion errors" by Tom Charlton [1]), and the development of artifacts and procedures to document this uncertainty. This paper will present the initial results of our experiments, and introduce methods to evaluate these data for both performance and diagnostic information.

Key Words: *Multi Sensor CMMs, Opto-mechanical hole plate.*

PRESENT WORK

We have performed two types of measurements on the multisensor CMM at UNC Charlotte. The first measurements involve the comparison of individual features using different sensors; these are meant to expose errors in a similar manner to the B89 multi-tip tests. The second measurements utilize an artifact called an "opto-mechanical hole plate" shown in Figure 3, and are intended to reveal errors in the machine volume that may be different for the different sensors. This work builds on earlier work with our multi-sensor CMM [2] and on characterization of video sensing CMMs [3] conducted at UNC Charlotte.

Sensor offsets

In the first measurement setup, we calibrated the touch probe and vision sensor on a common setting ring, and then measured a different ring and a gage block at the other end of the measuring volume of the machine. The setting ring was in the lower left edge (0, $\frac{1}{2}$, 0) of the CMM volume, and the other artifacts were in the upper right edge (1, $\frac{1}{2}$, 1) of the measuring volume. This arrangement is shown in Figure 1. A common origin was set at the center of the setting ring, and the larger ring (Ring 2) was measured 5 times with each sensor.

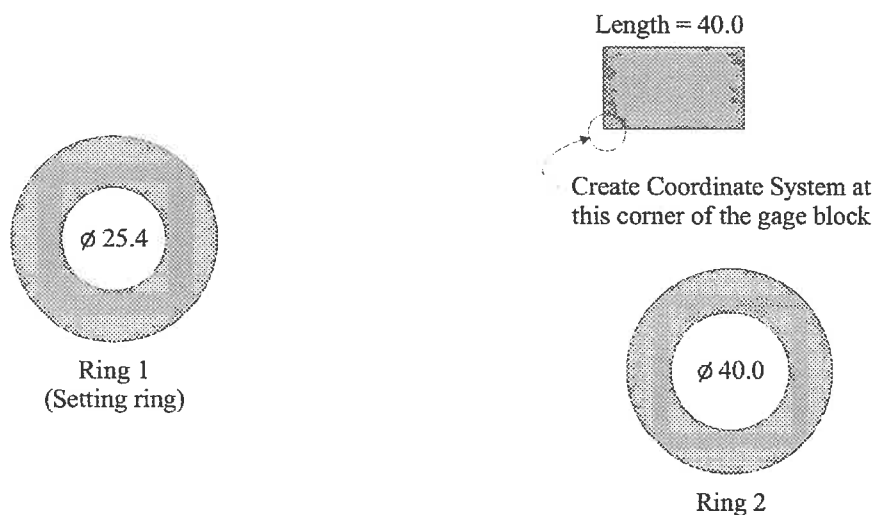


Figure 1: Artifact setup on multisensor CMM (gage block and Ring 2 are raised in Z)

The results of repeated measurements on the larger ring show some scatter in the center location of the ring for each of the two sensors, and systematic error between the sensors. To check that there was no excessive drift or hysteresis for either sensor, we returned to the setting ring after each measurement and reverified its location. The total scatter for both sensors was less than 3 micrometers in x and y, which leads us to conclude that there is some difference in the machine errors and compensation as we traverse the machine volume.

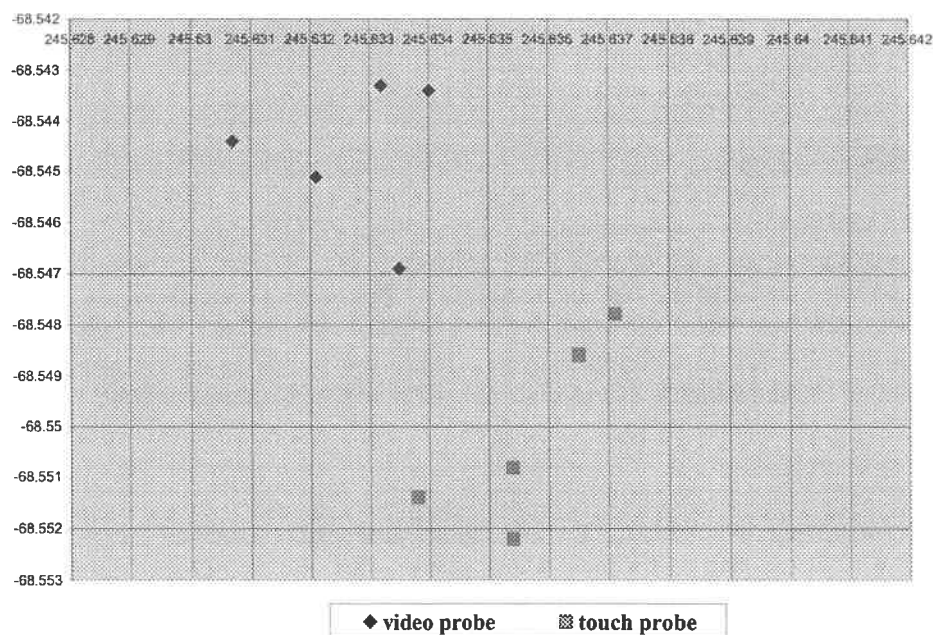


Figure 2: Comparison of Ring 2 center location measurements (1 micrometer grid).

Although not shown in this short abstract, we also found significant offset (over 20 μm) between measurements taken with the touch and video probes on the smooth gaging surface of the gage block. This is due to the highly reflective surface being measured in conjunction with a lack of experience with

various illumination techniques. While there may be better ways to measure this surface than we used, illumination remains a bit of a "black art" in video-sensing measurements.

Hole plate measurements

The next set of experiments used an opto-mechanical hole plate, provided by our colleagues at the Technical University of Denmark. This plate, shown in Figure 3, has a 100 μm thick stainless steel sheet captured between two heavier plates. The features measured in our experiments are the smaller through-holes which are spaced at a nominal dimension of 20 mm. Two features of this artifact that make it particularly useful for our project. The first is that the center sheet is both thin enough that the circular features are essentially two dimensional (facilitating video measurements) and thick enough that a touch probe can perform adequate contact measurements. The second nice feature is that the artifact can be "flipped" in the x- or y-direction, yielding simpler reversal mathematics than when repeatedly rotating the artifact.

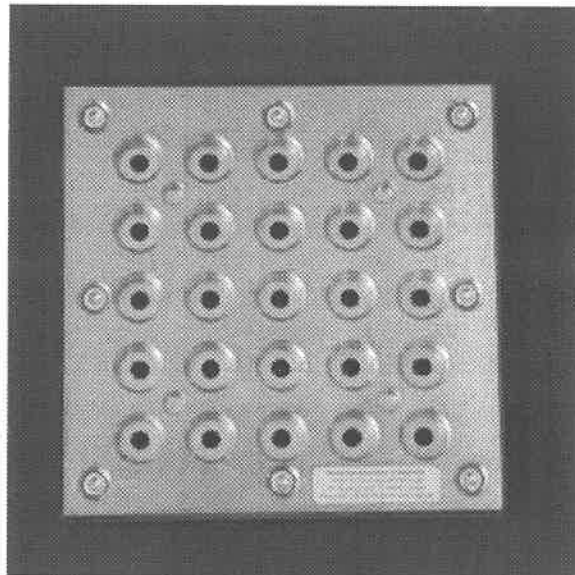


Figure 3: Opto-mechanical hole plate.

We have measured the hole plate in four orientations – listed in Table 1 – to permit the extraction of both CMM and artifact errors. The plate is positioned nominally in the center of the CMM volume, and to date we have measured it with a contact probe on our Leitz PMM to establish reference values for the hole locations, and with the video probe on our multisensor CMM.

Table 1. Positions of the Hole Plate

<i>No.</i>	<i>Description</i>
I	Start position
II	Plate flipped about the X-axis, reversing the y-coordinates relative to the start position.
III	Plate flipped about the Y-axis, reversing the x-coordinates relative to the start position.
IV	Plate rotated 180 degrees about its center, reversing both x- and y-coordinates.

The plot shown in Figure 4 is typical of the results we obtain. This plot compares the measurements taken in position I and position II. In every case, the origin and coordinate system of the plate is

constructed using the same physical features, so the extraction and separation of errors is slightly different than in the conventional reversals. In Figure 4, the deviations are shown magnified by 2000 times. This means that the 20 mm squares corresponding to the hole locations represent 10 μm when looking at the deviations between two measurements.

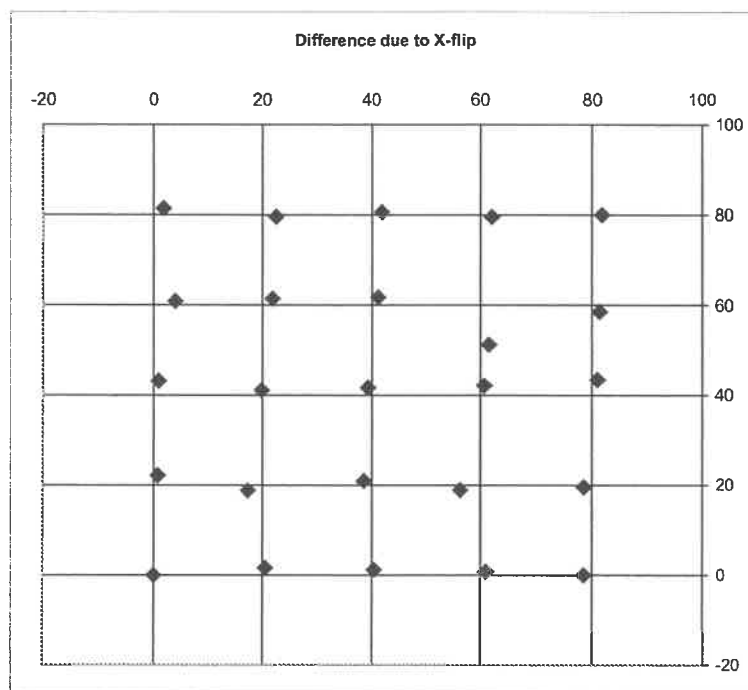


Figure 4. Comparison of measurements after reversal (see text for notes on scale).

Future Work

This work will soon include measurements on the multisensor CMM using *both* the video and touch probe, which will help us separate errors in the CMM due to sensor changes from uncorrected errors in the machine structure. In addition, comparison to other measuring machines will allow us to assess the range of errors that can be revealed with an artifact of this type.

Acknowledgements

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A New Method to Measure Runout of Motorized End-Milling Spindles at Very High Speed Rotations

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Abstract

The error motion of a machine tool spindle directly affects the dimensional accuracy of its machined parts. Spindle runout is a simplified indication for the spindle error motion. Traditional runout measurement techniques usually are applicable only when the spindle is at rest or at low rotational speeds. At very high speeds, such as 25,000 RPM or above, there are several factors, such as airflow, vibration, and difficulties in sensor fixturing, that could prevent proper measurements of the spindle runout. This paper introduces a new, practical method that can measure spindle runout at very high speed rotation. The spindle runout is determined by cutting marks produced by an end mill on a workpiece. The cutting marks are circular and their diameters are related to the spindle runout. The increase in diameters of these cutting marks at high speeds, with respect to the one by hand rotation, is related to the runouts at those speeds. Quantitative precision analysis was carried out to confirm the accuracy, repeatability, and resolution of this new runout measurement technique. This technique was applied to determine the quantitative relationships between the spindle runout and the spindle speed, the level of unbalanced, and the front bearing preload. The results confirm that the spindle runout increases as the spindle speed increases mainly due to the centrifugal forces generated by the unbalanced mass, which is consistent with the prediction by a comprehensive spindle thermal-mechanical-dynamic model previously developed by the authors. One way to compensate for the effect of unbalanced mass is by increasing the spindle stiffness. Experimental results confirm that a higher front bearing preload can render the spindle stiffer, thus reducing the spindle runout. This new and practical technique for spindle runout measurement is an effective tool for verifying actual running accuracy of spindles at their actual operating speeds, which is highly valuable in quality assurance.

1. Proposed Runout Measurement Method

1.1 Cutting mark generation

A workpiece is held on a linear slide moving along the axial direction of the end mill. Before a cutting mark is generated, it is moved to within an arbitrary small distance from the tip of the end mill and remains there until the spindle temperature reaches its steady state. After the steady-state temperature is reached, which takes approximately 20-30 minutes, the workpiece is fed into the end mill to make a shallow cutting mark of 0.00048 in (12 μ m) in depth for one minute. The depth of cut must be very shallow and a very sharp edge cutter must be used. If the depth of cut is too deep, other effects such as self-excited vibration, and rapid wear of the tool, might affect the profile of the cutting marks.

1.2 Diameter measurement of cutting mark

A high-resolution edge detector machine, MicroVu, is used for measuring the diameter of the cutting marks. The resolution of the MicroVu can be changed to a finest resolution of

0.000005 in (0.0001 mm). After setting up a workpiece on the measurement table with a proper fixture, a profile is displayed on a monitor screen. An operator can control an edge detector cursor to determine the X-Y coordinate of a point at the cutting profile edge. After selecting three or more points, up to nine points, on the cutting profile, the diameter of a best-fit circle is automatically calculated.

1.3 Cutting marks and spindle runout

The correlation between the spindle runout and the cutting marks is described for an idealized case as follows. If the spindle is perfect, a circular cutting mark of diameter D_0 is generated. The diameter D_0 is equal to $2R$, where R is the radius of the end mill. As the spindle rotates at a higher speed, it demonstrates an error motion, shown as the small irregular circular profile. A best-fit circle of radius, e , is used to represent the error motion profile. Due to this spindle error motion, a larger diameter cutting mark is generated. Its diameter is represented by a corresponding best-fit circle, measured to be $2(R+e)$. As a result, the difference between D_1 and D_0 is

$$\delta_r = D_1 - D_0 = 2e \quad (1)$$

, which is equivalent to the conventional runout measurement as two times of the error motion radius.

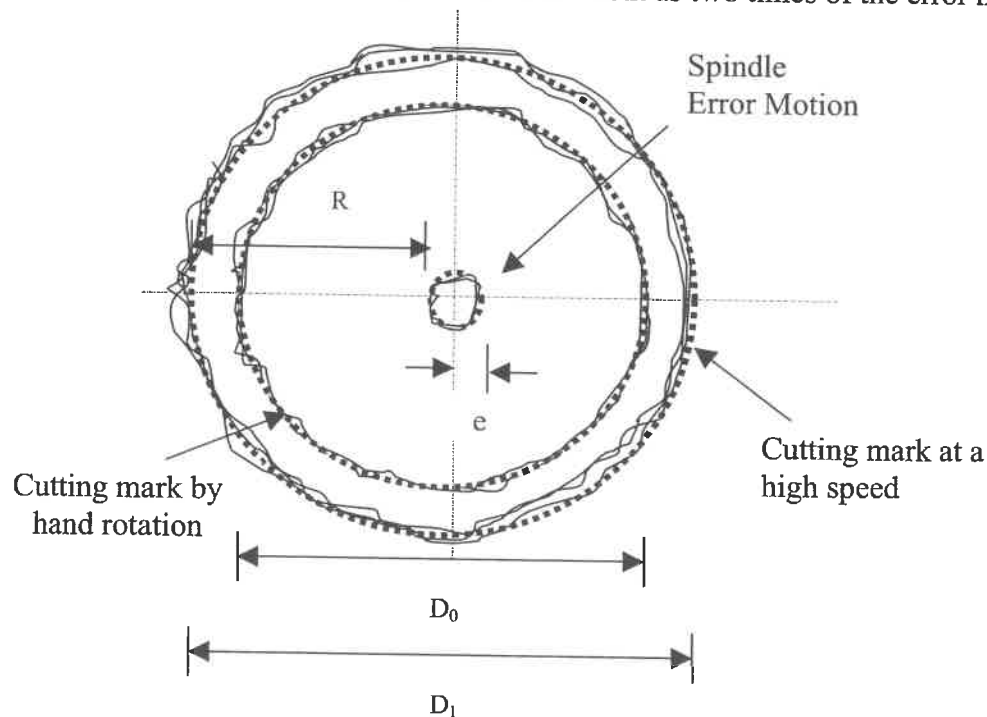


Figure (1): Cutting mark at 0 rpm and at N rpm

In practice, a best-fit circle of the cutting mark generated by hand rotation is used in the place of the perfect cutting mark of Fig (1), while the outer edge profile is used to represent the cutting mark at high speed. The above analysis establishes a correlation between δ_r and the spindle runout.

2. Precision Analysis of the Proposed Spindle Runout Measurement

2.1 Spindle Runout Measurement System

A measurement system consists of several components with its own inaccuracy. Error due to each measurement component contributes to the repeatability of the overall measurement. By performing measurement precision analysis, the dominant source of error can be determined. The overall spindle runout measurement (δ_r), as described below in a general form:

$$\delta_r \pm \Delta\delta_r = f(u_1 \pm \Delta u_1, u_2 \pm \Delta u_2, u_3 \pm \Delta u_3, u_4 \pm \Delta u_4) \quad (2)$$

Where δ_r is the overall computed spindle runout; $\Delta\delta_r$ is the overall repeatability error of the spindle runout measurement; u_1 is related to the cutting process; Δu_1 is the error due to the cutting process; u_2 is related to the linear slides; Δu_2 is the error due to the linear slides; u_3 is related to the diameter measurement; Δu_3 is the error due the diameter measurement; u_4 is related to temperature; Δu_4 is the error due to temperature.

Equation (2) can be expanded via the Taylor series and by neglecting higher order term (H.O.T.), the value of $\Delta\delta_r$ can be estimated based on a worst case scenario, shown as Equation (3), and a best case scenario as Equation (4).

$$(\Delta\delta_r)_{\max} = |\Delta\delta_1| + |\Delta\delta_2| + |\Delta\delta_3| + |\Delta\delta_4| \quad (3)$$

$$(\Delta\delta_r)_{\min} = \sqrt{(\Delta\delta_1)^2 + (\Delta\delta_2)^2 + (\Delta\delta_3)^2 + (\Delta\delta_4)^2} \quad (4)$$

,where $\Delta\delta_n$ is the overall repeatability error due to u_n ; for $n=1,2,3,4$.

Equation 4 and 5 provides an estimation of the overall measurement repeatability error.

2.2 Determining Overall Measurement Repeatability Error ($\Delta\delta_r$)

The overall measurement repeatability error of the spindle runout, $\Delta\delta_r$, in Equation (2) can be obtained directly by checking the repeatability of several cutting marks. For each of the cutting mark, five diameter measurements were obtained using the nine-point routine. The grand average (D) and the overall standard deviation(σ) are determined. The repeatability error of δ_r can be determined as

$$\Delta\delta_r = \pm 3\sqrt{\text{var}(D_1 - D_0)} = \pm 3\sqrt{\sigma_1^2 + \sigma_0^2} \quad (7)$$

,where D_1, σ_1 and D_0, σ_0 are the grand average and standard deviation of cutting marks at high and hand rotational speed, respectively.

2.3 Error Component Analysis

With the overall measurement repeatability error, $\Delta\delta_r$, determined, four individual error sources, $\Delta\delta_1, \Delta\delta_2, \Delta\delta_3$, and $\Delta\delta_4$, in Equation (3) are then investigated and compare to, $\Delta\delta_r$, for determining their relative significance.

Finally, all four-error sources are summarized in Table (1). According to the relationship between the error sources and the overall error defined by Equation (4) and (5), it is found that the dominant error source is the routine used to calculate the diameter of a cutting mark. As $\Delta\delta_2$ and $\Delta\delta_4$ are negligible, the error source due to the cutting mark process can be estimated to be ± 0.00008 according to Equation (4). Finally as indicated by Table (3) and (4), the overall repeatability can be improved by choosing a routine with more data points.

Table (1): Errors due to each effect

Effect	Errors	Error (in)
Generating a cut mark process	$\Delta\delta_1$	Unknown
Linear Slides	$\Delta\delta_2$	Negligible
MicroVu	$\Delta\delta_3$	± 0.00056
Temperature	$\Delta\delta_4$	Negligible
Overall repeatability	$\Delta\delta_r$	± 0.00064

3. Experimental Studies of Spindle Behaviors at high Speed Runout

At high-speed rotations, several phenomena, such as centrifugal forces, unbalanced mass effect, and thermally induced preload, can affect the runout of the spindle system. In this section, our proposed spindle runout measurement technique will be applied for studying these effects on the spindle runout at high-speed rotation.

The average diameters and the corresponding spindle runouts at six different spindle speeds: 0 rpm, 5,000 rpm, 10,000 rpm, 15,000 rpm, 20,000 rpm, and 25,000 rpm, are presented in Table (6) and plotted in Figure (8).

Table (6): Spindle runout at different spindle speeds up to 25,000 rpm

Speed (RPM)	Diameter (in)	Spindle Runout
0	0.4972 ± 0.0004	-
5000	0.4978 ± 0.0004	0.0006 ± 0.0005
10000	0.4984 ± 0.0002	0.0012 ± 0.0004
15000	0.4990 ± 0.0003	0.0018 ± 0.0005
20000	0.4992 ± 0.0002	0.0020 ± 0.0004
25000	0.4996 ± 0.0001	0.0024 ± 0.0004

The spindle runout at 0 rpm was measured by a capacitance probe.

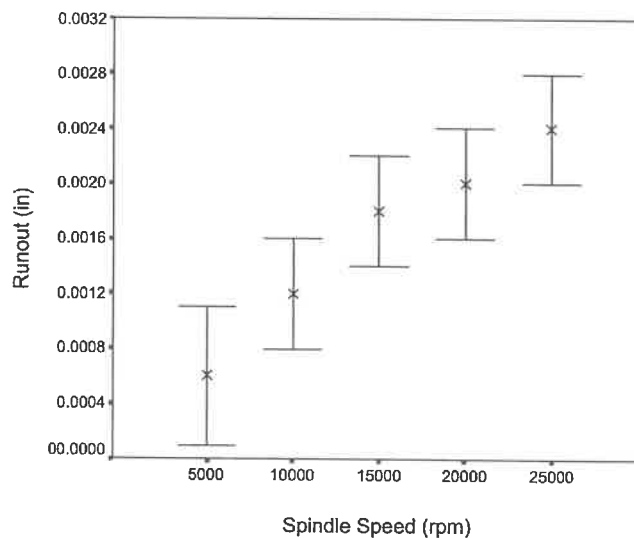


Figure (2): Runout plot of circle marks with respect to spindle speed

The results show that the runout is 0.0005 in at 0 rpm and it increases with the speed, becoming 0.0024 ± 0.0004 in at 25000 rpm.

Evaluation of Measurement Uncertainty in Gear Measuring Instruments by Using the Monte Carlo Simulation

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Introduction

Recently, the need to perform uncertainty estimation is increasing, and many examples of uncertainty estimation have been presented. However, most of the examples can be expressed by model equations, and very few examples of complex quantities (i.e. industrial quantities) have been given. Because the factors involved in uncertainties are intricately interconnected in industrial quantities, it is very difficult to calculate the sensitivity coefficients of these factors.

The same is true also for gear measurement. In Japan, the Japan Gear Manufacturers Association established a committee of gear measurement accuracy for investigating gear measurement traceability and calibrations. One of the purposes of the committee is to estimate the uncertainty of gear shape measurement.

The geometry of a gear measurement machine, which is an industrial quantity measurement machine, is complex, and it is very difficult to estimate the uncertainty by applying the law of propagation of uncertainty which is defined by the Guide to the Expression of Uncertainty in Measurement (GUM). In this study, we estimate the geometrical uncertainties of gear measurement by using the Monte-Carlo simulation.

Gear measurement machine

The architecture of a gear measurement machine is shown in Fig.1. A linear slide is parallel with the y (tangent)-axis. A probe is attached to the linear slide. When the gear shape is measured, the probe is located at a point in the direction of the x (radial)-axis distant from the origin by the radius of a base circle. If the gear is turned, the tooth of the gear pushes the probe, and the length of probe movement is measured as y . There is a linear relation between the length of probe movement y and the angle of rotation θ .

$$y=r\theta$$

Here, r means radius of base circle. The difference between the measured value and ideal value is the profile deviation.

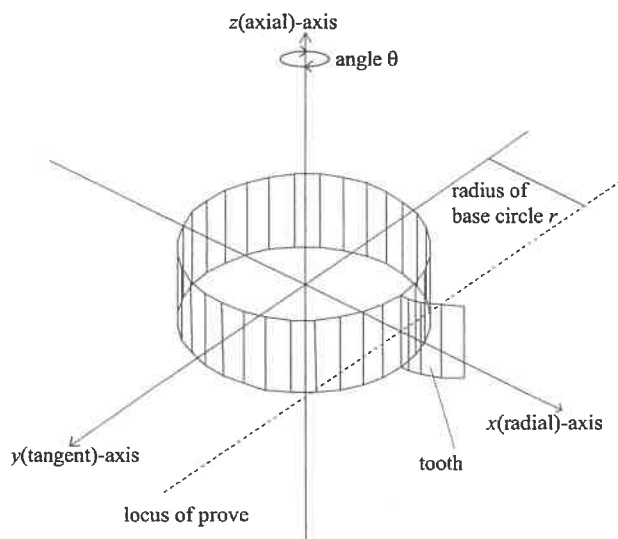


Fig.1: The architecture of gear measurement machine

Sources of geometrical uncertainties in gear measurement

Because the geometry of gear measurement machine is complex, it is very difficult to estimate total geometrical uncertainty. However, it is possible to estimate individual geometrical uncertainties. The sources of individual uncertainties which are considered in this study are shown in Fig.2 and Fig.3.

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⁷ Ogasawara Precision Hob Laboratory Ltd.

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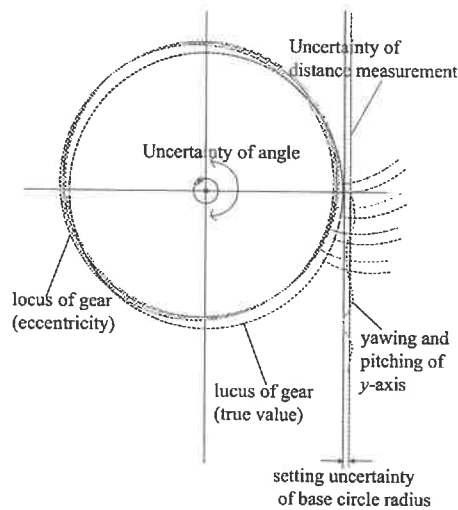


Fig.2: Source of uncertainties (x-y plain)

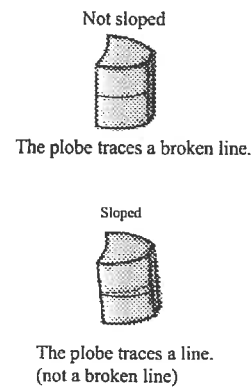


Fig.3: Source of uncertainties (z-axis is considered)

Table1: Individual uncertainties

Source of uncertainties	Value
Eccentricity (x-axis direction)	0.51 μ m
Eccentricity (y-axis direction)	0.51 μ m
Origin setting error (x-axis direction)	0.32 μ m
Origin setting error (y-axis direction)	0.12 μ m
Length measurement uncertainty (x-axis direction)	0.3 μ m
Length measurement uncertainty (y-axis direction)	0.0296 μ m
Length measurement uncertainty (z-axis direction)	0.3 μ m
Pitching of y-axis	3.73 μ m
Yawing of y-axis	0.684 μ m
Angle of x-axis for the reference plane	0.001 μ m/100mm
Angle of x-axis against the reference plane	0.001 μ m/100mm
Angle of y-axis for the reference plane	0.000245rad
Angle of y-axis against the reference plane	0.000118rad
Uncertainty of an angle of rotation	0.00000242rad
Slope of axel (x-axis direction)	0.0000060rad
Slope of axel (y-axis direction)	0.0000154rad

The preceding individual uncertainties were measured at the University of Electro-Communications and Osaka Seimitsu Kikai Co., Ltd¹⁾. The results of the measurements are shown in Table1²⁾. These results were substituted by a Monte-Carlo simulator, and we thereby obtained an uncertainty of gear tooth measurement.

Monte-Carlo simulation and result

Measurement and simulation conditions are as defined below,

- Rotation angle of gear.....37deg.
- Number of partitions of rotation angle.....45
- Radius of base circle.....49.4mm
- Model of tooth.....300000 points per line, 0.1mm pitch in the z-axis direction.
- Repetitions.....1000

Individual uncertainties shown in Table I are limited values except the measurement uncertainty (y-axis direction). The distributions of individual uncertainties except the length measurement uncertainty (y-axis direction) are assumed to be rectangular distributions, and the distribution of the length measurement uncertainty (y-axis direction) is assumed to be a normal distribution. The detail of the simulation is given below. First, a virtual machine of the gear measurement machine is programmed. This virtual machine has factors of geometrical uncertainties which are the same as the factors of a real gear measurement machine, and random values which conform to the fixed distributions are generated. The random values are inputted into the virtual machine, and we obtain an output which represents the measurement results including whole geometrical uncertainties. If this procedure is repeated, we obtain the distribution of measurement results, and we can calculate expanded uncertainties. The detail of the estimation uncertainty flowchart is shown in Fig. 4. Here, the number of repetition is 1000 times, and the results are arranged by size starting with the smallest. The 950th value from this size represents expanded uncertainty as the expanded uncertainty is an interval having a level of confidence of approximately 95%.

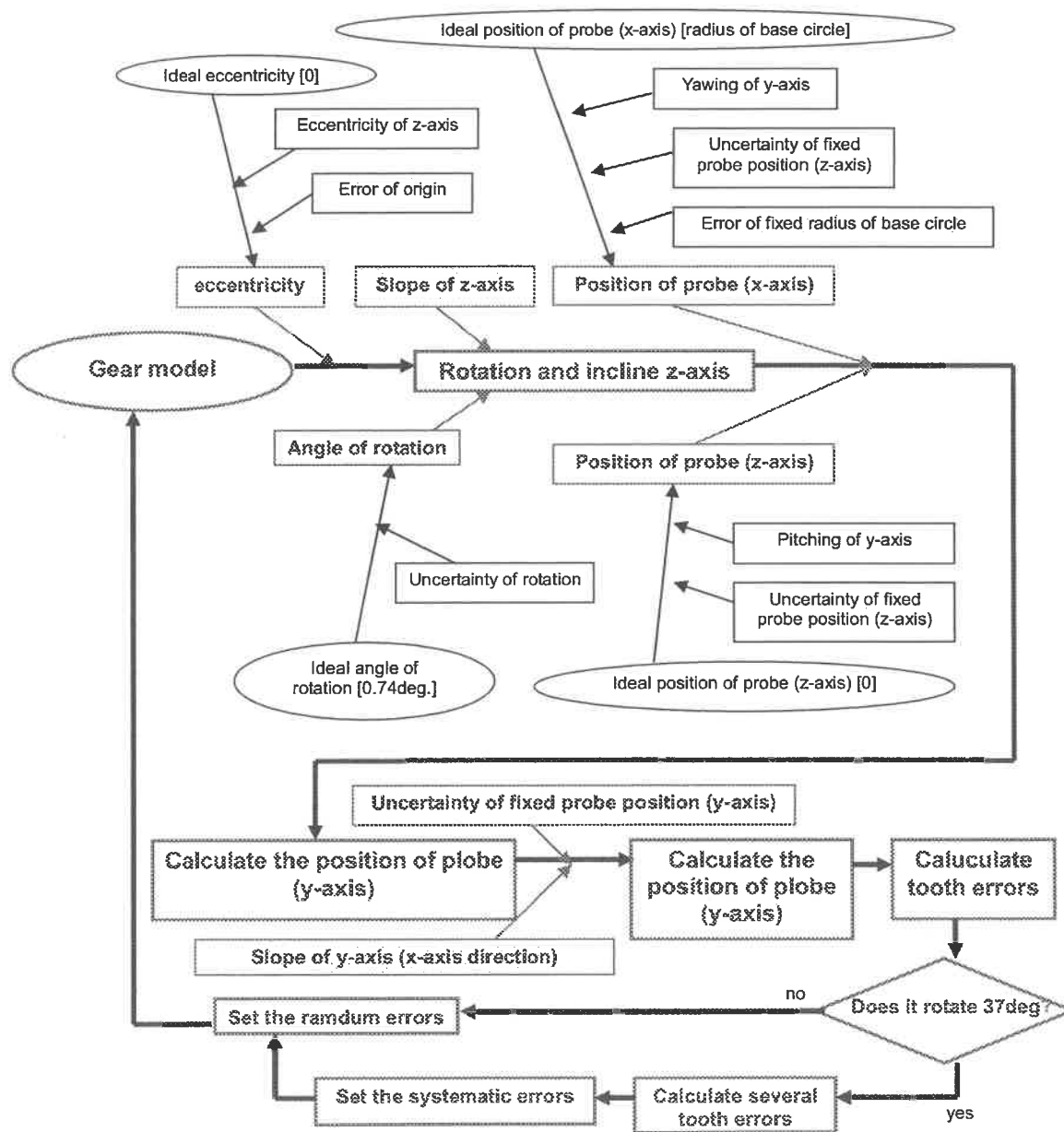


Fig.4: Flowchart of the simulation

The results of the simulation are shown below. We estimate three uncertainties of profile deviations which are prescribed by ISO 1328-1:1995, namely total profile deviations, profile form deviations and profile slope deviations. The results of estimating uncertainty are given below,

Measurement uncertainty of total profile deviations	0.73 μm
Measurement uncertainty of profile form deviations	0.43 μm
Measurement uncertainty of profile slope deviations	0.49 μm

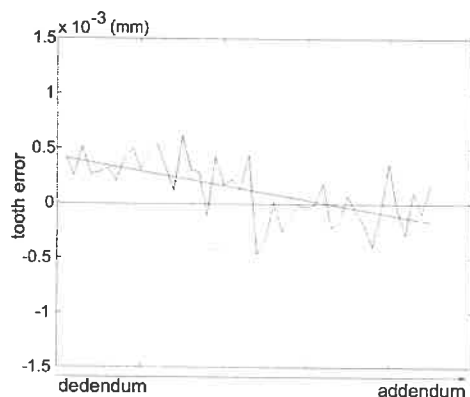


Fig.5: Example of result of simulation

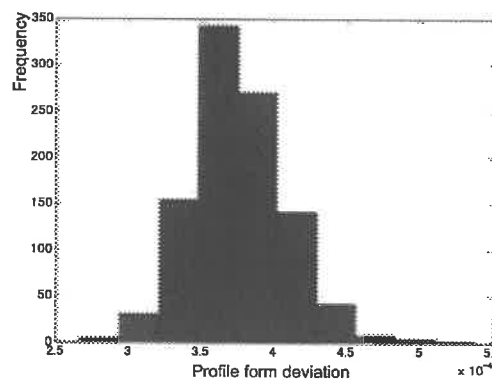


Fig.6: Histogram of profile form deviation

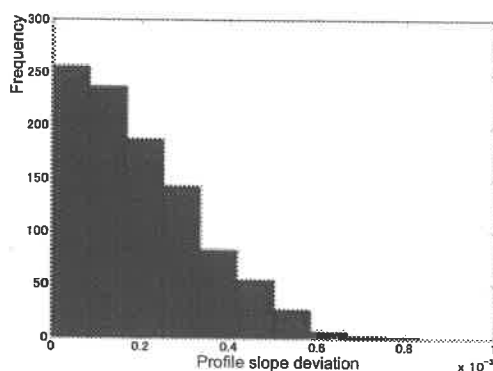


Fig.7: Histogram of profile slope deviation

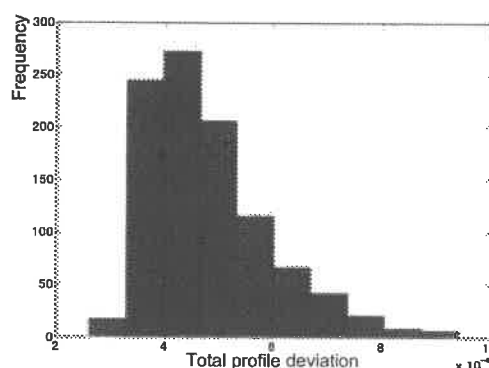


Fig.8: Histogram of total profile deviation

One of the results which was obtained from 1000 simulations is shown in Fig.5, the histogram of profile form deviation is shown in Fig.6, the histogram of profile slope deviation is shown in Fig.7 and the histogram of total profile deviation is shown in Fig.8.

The histogram of the profile form deviation is a normal distribution, but the histogram of the profile slope distribution decreases linearly. The reason is that the profile form deviation is a random error, and the profile slope deviation is a systematic error. The total profile deviation is a combined profile form deviation with profile slope deviation, which is an asymmetrical distribution. Generally, in an asymmetrical distribution, it is difficult to calculate expanded uncertainty, but in this case, it is very easy to find the 950th value which represents expanded uncertainty.

Conclusions

This study estimates the expanded uncertainty of a measuring gear tooth profile which is a very complex value. However, there is the possibility of overestimation in order to apply the distribution of a limited value to a rectangular distribution. This point must be improved. In the future, we will structure the simulation for helix deviations and pitch deviations as well as profile deviations.

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Surface Metrology Software Variability in Two-Dimensional Measurements

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1. Introduction

A range of surface texture measurement instruments is available in the market place. Most of the measurement instruments are microcomputer-based systems that contain their own surface analysis software to evaluate measured roughness profiles. After a measurement is taken, data points representing a surface profile are filtered to remove unwanted wavelengths due to instrument noise, tool marks, or chatter, etc. Surface texture parameters are calculated based on the filtered profile and used to characterize the surface profile. Over a hundred parameters have been defined for industrial use, and many of these appear in national standards as well. Some standards organization such as ISO (International Organization of Standardization) and ASME (American Society of Mechanical Engineers) have different surface texture parameter definitions and evaluation procedures. Thus, different implementations can yield differences in results. This paper describes the results of a surface finish parameter round-robin variability investigation among commercial software packages for surface texture analysis. The parameters included in the study are R_a , R_q , R_{sk} , R_{ku} , R_z , R_{dq} , R_p , R_v , R_t , and R_{sm} .

In the past, ASME Committee B46 (ASME B46) on the Classification and Designation of Surface Qualities conducted a surface parameter algorithm round-robin to ascertain the relative agreement between various surface metrology software packages [1]. The surface parameter algorithm round-robin showed that results for some surface parameters did not agree with one another. However, it did not mention which standards should be used in the round robin. The choice of standards, ISO or ASME, will influence measurement results, because of differences in the mathematical equations used in each case. This paper describes a surface finish parameter round-robin variability investigation among software packages. The investigation was conducted on surface parameter calculations based on both ISO and ASME B46 Standards [2][3][4]. The purposes of this investigation are to show surface metrology software variability in two-dimensional measurements and to show how surface parameters calculated from the two standards give different results for surface parameter calculations.

2. Commercial Software Packages Inter-Comparison and Results

A set of 2D profiles with 0.5 μm spacing and 4 mm trace length from the previous ASME B46 investigation and five commercial software packages were used in the investigation. The 2D profile data set included a numerically generate uncorrelated random profile, sinusoidal profile, square wave and impulse, and three traces from ASME B46 type-D roughness specimens, which were measured, preprocessed, and scaled to the (40 to 50) μm R_a range [4]. The comparison was based on roughness profiles filtered using a Gaussian filter at 0.8 mm cutoff. The surface parameters calculated by commercial software packages were compared to one another and also to National Institute of Standards and Technology (NIST) calculated surface parameters.

Commercial packages have company-specific data handling formats. Some of them allow the user to import external data for analysis while some do not. Therefore, it is sometimes impossible to use a calculated roughness profile from one partner to characterize it with software from another partner, if the software does not have an ability to import and export surface profile data. In this paper, surface parameters calculated from five commercial software packages that have this ability are investigated. The ASME B46 data set was imported to commercial software packages to generate roughness profiles and calculate surface parameters. The roughness profiles were then imported to NIST Surface Metrology Algorithm Testing System for surface parameter calculation [1]. The surface parameters calculated by the commercial software packages were compared with NIST's calculations and the percentage differences are reported. The results are shown in figure 1.

Results:

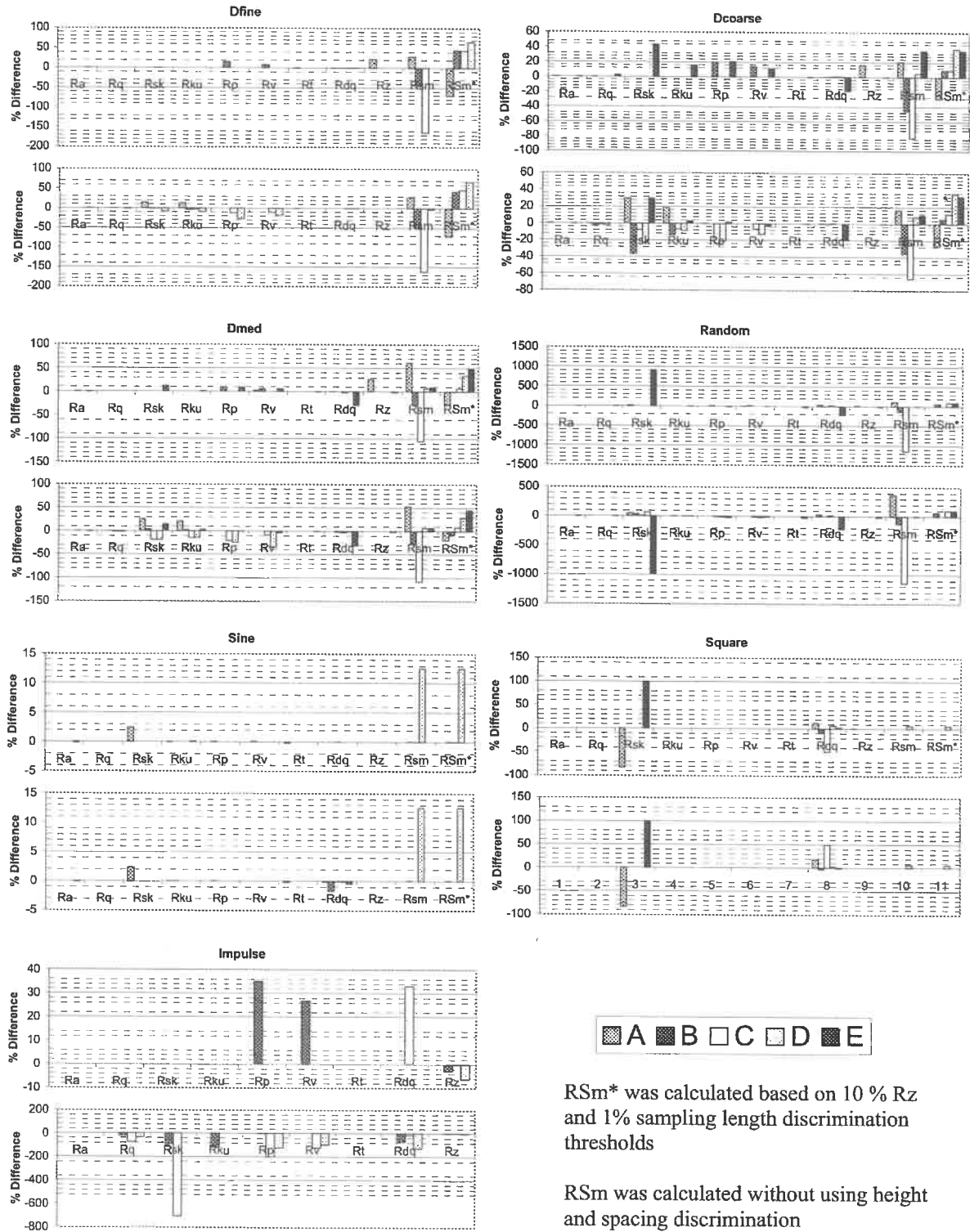


Figure 1. Results of the comparison among software packages for surface analysis

NIST calculated surface parameters used in this comparison were based on both B46 standards and ISO Standards. There are two plots for each surface profile. The first plot shows the comparison of surface parameters relative to NIST calculated surface parameters based on ASME B46 standards. The second one shows the comparison of calculated surface parameters relative to NIST calculated surface parameters based on ISO 4287 [2] and ISO 4288 [3]. For the commercial packages the standard serving as the basis for the calculations was the same for both graphs and in four cases was not known. ISO and ASME have different surface parameter definitions and evaluation procedures. ISO 4287-1997 defines some parameters within a sampling length and others within an evaluation length. ASME B46 defines most parameters within the evaluation length. To evaluate an entire profile length, ISO 4288 indicates that an average of the parameter estimates from all available sampling lengths are taken, and this parameter is called the average parameter estimate. The following describes conclusions drawn from figure 1.

- Software packages A, B, C, and D have very small percentage difference (0 %-2.4 %) with NIST's calculation for Ra , Rq , Rsk , Rku based on B46 Standards for most data set. The small percentage difference may be caused by different implementation (eg. float vs. double used to store numbers) although the same mathematical equations described in standards are likely used. For the calculations based on ISO, a higher percentage differences were observed. So it can be said that software packages A, B, C, and D agree with one another when they use B46 standards to calculate these parameters.
- RSm does not agree among commercial software packages A, B, C, D, and E for calculations based on both B46 and ISO Standards. For sinusoidal and square profile, the percentage differences are less compared with other profiles. This disagreement may be caused by height and spacing differences defined in ISO standards. Leach et al. [5] showed that the definition of the RSm parameter given in ISO 4287 standards was ambiguous, leading to the possibility of different algorithms for calculating RSm .
- Software B calculated Rp , Rv based on ISO 4287 and ISO 4288. On the other hand, software A, C, D, and E calculated these parameters based on B46 standards and they agree with each other. Software A has the ability to calculate surface parameters based on both ISO and ASME B46 standards.
- Software A, B, C, D, and E agree on the Rz calculation.
- Software A, B, and D agree on the Rdq calculation.

- For calculation of the arithmetic mean deviation parameter (Ra), ISO and ASME B46 have the same value. This can be proved by the following:

For ASME B46, $Ra = \frac{1}{L} \int_0^L |z(x)| dx$, where L is the evaluation length and $Z(x)$ is height of the profile at position x .

For the average Ra estimate defined in ISO 4287 and ISO 4288

$$Ra = \frac{1}{n} \left(\frac{1}{Lr} \int_0^{Lr} |z(x)| dx + \frac{1}{Lr} \int_{Lr}^{2Lr} |z(x)| dx + \dots + \frac{1}{Lr} \int_{(n-1)Lr}^{nLr} |z(x)| dx \right) = \frac{1}{L} \int_0^L |z(x)| dx \quad (1)$$

Where Lr is a sampling length, n is the number of the sampling length and L is equal to nLr

- Rp and Rv defined in ISO 4287-1997 are calculated based on an average of the parameter estimates from all available sampling lengths. ASME B46 defines these parameters based on the evaluation length. So it can be said that Rp and Rv of ASME B46 have greater or equal value to those defined in ISO. This can be shown in figure 1.
- Rt of ASME B46 is equal to Rt defined in ISO 4287-1997 since both define Rt based on the evaluation length.
- Rz of ASME B46 is equal to Rz defined in ISO 4287-1997 since both define Rz based on the sampling length.
- Rq of ASME B46 is higher than Rq of ISO. This is shown by the following:

ASME B46 defines Rq as

$$Rq = \sqrt{\frac{1}{L} \int_0^L z^2(x) dx} = \sqrt{\frac{1}{nLr} \left(\int_0^{Lr} z^2(x) dx + \int_{Lr}^{2Lr} z^2(x) dx + \dots + \int_{(n-1)Lr}^{nLr} z^2(x) dx \right)} \quad (2)$$

ISO defines Rq estimated as

$$Rq = \frac{1}{n} \left(\sqrt{\frac{1}{Lr} \int_0^{Lr} z^2(x) dx} + \sqrt{\frac{1}{Lr} \int_{Lr}^{2Lr} z^2(x) dx} + \dots + \sqrt{\frac{1}{Lr} \int_{(n-1)Lr}^{nLr} z^2(x) dx} \right) \quad (3)$$

$$\text{Let } A_1 = \sqrt{\frac{1}{Lr} \int_0^{Lr} z^2(x) dx}, A_2 = \sqrt{\frac{1}{Lr} \int_{Lr}^{2Lr} z^2(x) dx}, \dots, A_n = \sqrt{\frac{1}{Lr} \int_{(n-1)Lr}^{nLr} z^2(x) dx}$$

Substitute $A_1, A_2, \dots, A_n$ back to equation (2) and (3)

$$R_{qB46} = \sqrt{\frac{A_1 + A_2 + \dots + A_n}{n}} \quad (4)$$

$$R_{qISO} = \frac{\sqrt{A_1} + \sqrt{A_2} + \dots + \sqrt{A_n}}{n} \quad (5)$$

Using Arithmetic-Geometric Mean Inequality ($a/2 + b/2 \geq a^{1/2}b^{1/2}$, $a \geq 0, b \geq 0$), it can be proved that R_{qB46} (Equation 4) is greater than or equal R_{qISO} (Equation 5). Using the same above proof for Rdq , it can also be seen that Rdq of B46 is greater than or equal to Rdq of ISO.

- Rku and Rsk depends on Rq value; they will have higher value if Rq is small. If Rku and Rsk are calculated based on the average Rq defined in ISO 4287 and ISO 4288, they will have higher or equal value than Rku and Rsk calculated based on the ASME B46 standard.

3. Conclusion

- Software is a primary contributor to variability in 2D surface measurement.
- A standards working group should be established to help ensure traceability in algorithm development by providing and maintaining a set of master algorithms and data sets for companies, universities, and instrument manufacturers to compare and validate their surface analysis systems.
- Commercial software manufacturers should specify clearly how surface parameters are implemented and which standards are used in their software.

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Large field-of-view scanning white-light interferometers

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1. Introduction

Scanning white-light interference (SWLI) microscopes routinely measure machined parts in industry, using broadband light, electronic detection, precision mechanical scanners and the principles of low-coherence interferometry to characterize surface form and roughness. Benefits of these instruments include their high Gauge Repeatability and Reproducibility (GR&R) capability, the large number of measurement points acquired in a small amount of time and the non-contact nature of the method.

Although there are many forms of SWLI microscope available, the maximum size of the parts they can measure is usually on the order of 5-10 mm (1X objectives). In this paper, we look at different implementations of large field-of-view (FOV) interference microscopes in the context of the measurement of machined surfaces. The extension of white-light interferometry to large objects is straightforward when surface finish is smooth. However, rough surface finishes tend to render the measurement process increasingly difficult as part size increases.¹ We describe these limiting effects, how they can be mitigated and show examples of measurement on production machined parts. An example tool is a par-focal and par-fringe, dual-objective interference microscope² used for form and roughness measurements on parts that can be as large as 40 mm. Finally, we briefly describe an experimental interferometer measuring rough parts as wide as 200 mm.

2. Optical profiling of large rough parts – Why is it difficult?

Low magnification optical systems tend to have low numerical aperture (NA), that is, they capture light scattered by the surface in a fairly reduced solid angle compared to higher-magnification systems. The low NA also restricts the lateral resolution of the instrument. In practice, most instrument manufacturers tend to match the lateral resolution to the footprint of a camera pixel at the object surface. However, because the lateral resolution spot is fairly large at low magnifications (30 microns for a 0.02NA objective) the range of random surface heights present within one spot can become comparable to the wavelength of light when measuring rough parts. This is the source of the so-called speckle phenomenon. In practice, each different illumination direction or wavelength generates a slightly different random intensity pattern at the detector, or speckle. Because of the random nature of this phenomenon the phase of the interferogram becomes meaningless and one no longer observes the macroscopic continuous interference fringes seen on smooth objects. However, white-light interferometry still detects regions of high-modulation interference, thus allowing surface profiling based on the presence or absence of interference. This is the domain of the coherence radar.³ The difference between the smooth and rough regimes is illustrated in Figure 1 where we show the interference signals detected at 9 neighboring pixels using an experimental 30-mm FOV white-light interferometer. We observe loss of fringe contrast,

random lateral displacements of the peak interference signals and distortion of the signal envelope in the rough regime.

These phenomena have been studied extensively at the University of Erlangen both theoretically and experimentally. The key conclusions can be found in the published literature.⁴ Among these, an important observation is that the noise of a surface height map generated using the location of peak interference contrast is directly proportional to the local surface height standard deviation, even though the height map itself is not representative of the actual microscopic shape of the surface. This property immediately sets a lower limit on the repeatability of the measurement. This limitation can be overcome by increasing the wavelength of light used for the measurement (i.e. to 5 or 10 μm), but that is not always a practical solution.

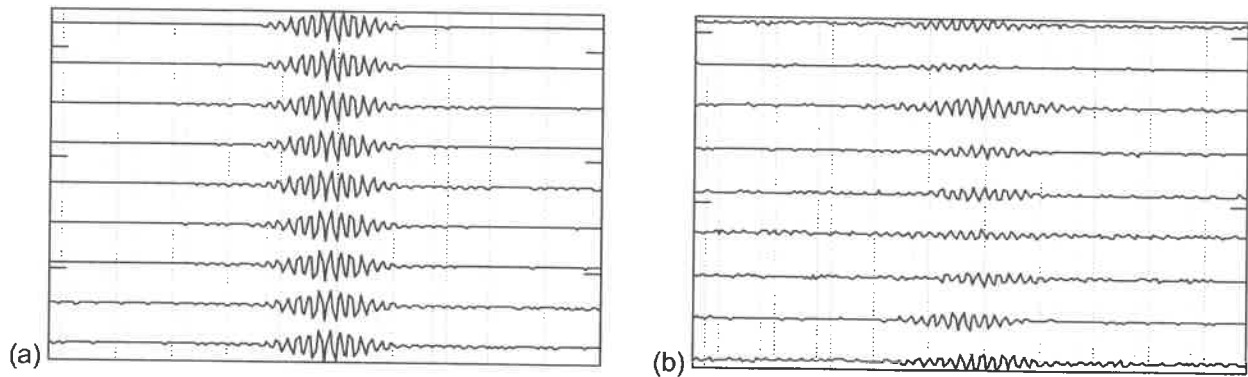


Figure 1 Comparison of white-light interference signals measured on a smooth (a) and rough (b) surface, using a 30-mm FOV white-light interferometer.

3. Instrument optimization

Keeping in mind the rather fundamental limitations outlined above, it remains necessary to optimize a white-light interferometer design to maximize the quality of the signal used for the measurement of rough surfaces. In practice, the transition from the smooth regime to coherence radar requires optimization of such system parameters as the source dimension, imaging system lateral resolution, scattered-light collection, source spectral width, reference mirror reflectivity, dispersion correction, etc. We can briefly describe the effect of the most influential factors and suggest optimal settings derived from our experiments on prototype large-FOV systems.

One of the most important parameters is the NA of the optical system. Large NA objectives capture more scattered light, which is an important benefit when measuring parts that scatter light in the entire half-space. Also, higher NA implies higher lateral resolution, which creates a beneficial speckle averaging effect at each pixel. As the number of speckles per pixel increases the contrast drops but the increased object amplitude at the detector compensates for this and the net effect is a relatively constant signal modulation. However, the practical benefit is that speckle averaging causes more pixels to detect a useable signal.

Ideally, the source should be as small as possible since distant source points generate different (also called decorrelated) speckle patterns. However, practical considerations when using broadband sources such as halogen lamps or LED impose finite source dimensions. Fortunately, the effect is not too

rapid and we observe a fairly linear contrast loss with source size, with a maximum 25% loss (for the rough parts used for this study) when the image of the source fills the pupil of the microscope objective.

A wide source spectral width produces a narrow signal envelope, increasing the interferometer resolution, while at the same time increasing the available amount of light. However, widely spaced wavelengths generate decorrelated speckles, resulting again in contrast loss. Experimental compromises are fairly dependent on the actual part roughness. However, a starting point consists in a full-width-at-half-maximum to central-wavelength ratio of about 15%.

Finally, because the amount of light scattered by the object in the instrument's pupil can be very weak compared to the reflection of the reference mirror, it is useful to decrease the reference surface reflectivity. Because the dynamic range to be covered can be quite large the ideal is an objective having adjustable reflectivity. We observed significant fringe contrast improvements when dropping the reflectivity of reference surfaces down to 4% and even below 1%. However, as the reference reflectivity is better matched to that of the rough surface the total amount of light captured by the instrument can drop dramatically. In practice, the optimum reflectivity is the one that maximizes detected signal modulation when the source is running at its highest output power. A consequence is that the optics need to be properly anti-reflection coated so as not to introduce a strong background intensity at the detector.

As a conclusion to this experimental investigation we ran a GRR on production lapped parts. The R_a roughness of these surfaces is close to $0.3\text{ }\mu\text{m}$, with a very short roughness correlation length, resulting in a very low apparent reflectivity. The one-standard deviation GR&R for flatness measurement is less than $0.08\text{ }\mu\text{m}$, a result consistent with meeting 10% gauge on a $4\text{-}\mu\text{m}$ flatness tolerance. These results were obtained after applying a 13×13 low-pass filter to the height data in order to emulate the lateral resolution of an infrared interferometer using a $10\text{-}\mu\text{m}$ broadband source.

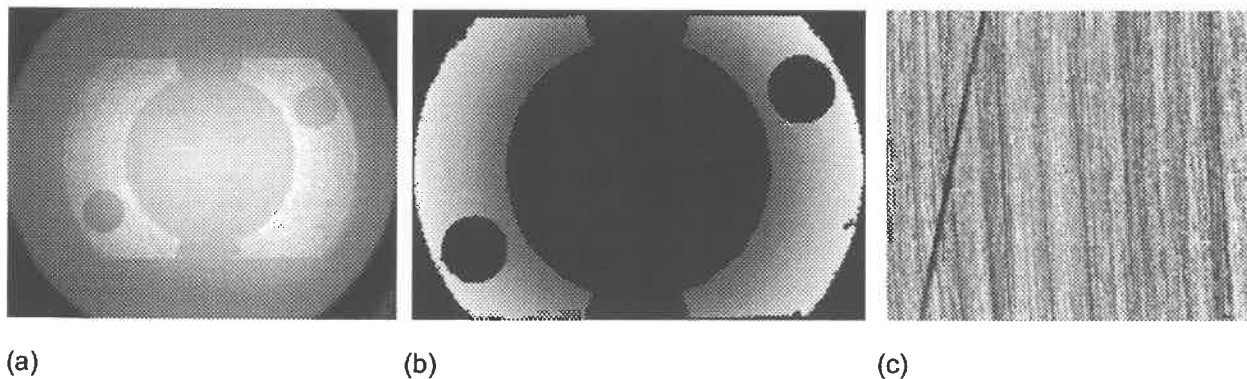


Figure 2 (a) 30-mm diameter part within the FOV of the 0.15X objective. (b) Height map of the surface. The part has a slight spherical shape creating an $8\text{-}\mu\text{m}$ sag. (c) Height map of a $0.7 \times 0.5\text{ mm}$ subregion captured with the 10X objective for roughness measurement. R_a is about $0.2\text{ }\mu\text{m}$ for this surface.

4. Example of a tool for the combined measurement of form and roughness

We have developed an instrument to exploit this new large-FOV capability by integrating form and roughness measurement in a package, combining two objectives on a two-position slide: a medium-magnification objective for local roughness characterization with a very-low-magnification objective for

form measurement. The system uses polarization elements to eliminate the light scattered by optical components in the microscope objectives, a source of contrast reduction when measuring rough parts.

The respective FOV covered are 0.70×0.53 mm (about 10X system magnification) and 42.7×32.0 mm (about 0.15X system magnification). So far this tool has been applied to the characterization of machined parts with roughness as large as $1.4 \mu\text{m Rz}$, while it routinely measures on the factory floor ground parts having an Rz roughness on the order of $0.2 \mu\text{m}$. Figure 2a shows a 30-mm diameter part within the FOV of the 0.15X objective. The corresponding measured form is shown in Figure 2b with a PV departure of about $8 \mu\text{m}$. The 10X objective allows capturing a magnified subregion for roughness characterization. The resulting height map is shown in Figure 2c. The Ra roughness is on the order of $0.2 \mu\text{m}$ for this particular surface finish.

5. Non-telecentric imaging – Very large aperture white-light interferometer

The interferometers studied so far are all based on telecentric objectives, which can indiscriminately measure smooth and rough surfaces. If one only intends measuring rough objects it becomes possible to use non-telecentric optics and perform measurements using a rough reference surface. This technique has been successfully demonstrated at the University of Erlangen where an aluminum casting about 200-mm wide was measured with scanning white-light interferometry using LED arrays for the illumination. Figure 3 at right shows the measured height map.

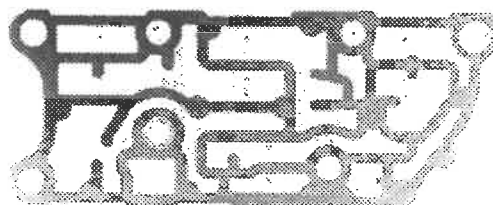


Figure 3 200-mm casting measured with SWLI. The flatness deviation is on the order of $30 \mu\text{m}$.

6. Concluding remarks

As optical profilers continue to improve they become more attractive metrology solutions for a number of industries, including the machined parts industry where a wide range of surface finishes are the norm. The addition of large FOV capabilities to current and new instruments, based on a better understanding of the physical phenomena encountered in the rough measurement regime, is a step forward in providing flexible, production-line ready tools to these industries for combined and easily automated form and roughness measurement.

7. References and notes

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ACCURACY ASSESSMENT FOR A LATERAL SHEARING INTERFEROMETER

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Introduction

For precision surface form error measurement under machining conditions, a new interferometer is proposed based on the shearing interference principle with a self-reference property. The interferometer is simple and compact in structure. It has a complete common optical path and is based on polarized phase shifting interference. It is less sensitive to vibration and environmental disturbances and is suitable for precision surface form aberration measurement.

The principle of the interferometer is introduced and its simplified optical model is presented. Several error factors of the interferometer are investigated. The results show that the accuracy of the proposed interferometer is less than $\lambda/80$. To confirm the results, experimental tests were conducted on the prototype of the interferometer.

Working Principle

The proposed shearing interferometer is shown schematically in Fig. 1. A single plate SP is the specially designed shear generator made of calcite crystal. By a laser source, a spatial filter and a collimated lens L_1 , a standard plane wave is produced. The plane wave incidents onto the test surface and is reflected back as an approximate plane wave whose wave front shape is double the test surface aberration. The reflected plane wave with the surface aberration information passes back through L_1 and another collimated lens L_2 and incidents into the shear generator SP. In SP, it breaks down into two perpendicularly polarized waves with different refraction indexes. When they come out from the plate,

the two waves have parallel propagation directions and the same aberration information of the test surface, with a little amount of shift between them. After they pass through a quarter wave plate Q_3 and a polarizer P_2 , they interfere with each other, thus shearing interference happens. By CCD, the interference pattern corresponding to shearing is obtained. Q_3 and P_2 consist of a polarized phase shifting device for multi-step phase shifting. In the phase shifting device, when P_2 is rotated to change its polarization direction by an angle α , a 2α phase shifting of interferometric phase can be produced. The obtained phase shifting shearing interference patterns

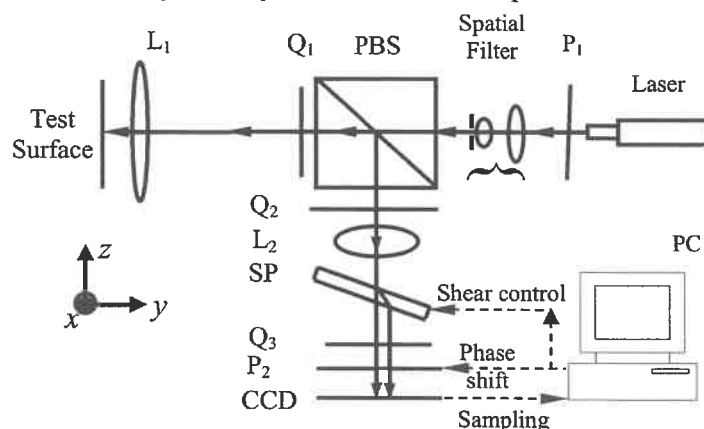


Fig. 1 Work principle of the interferometer

are sampled and analyzed to obtain shearing interferometric phases. Then by wave front reconstruction algorithm, the description of the wave front, hence the test surface topography aberration, is achieved.

Error Analysis

For the interferometer, several error factors are analyzed.

1) System error from distortions of optical components. In the built shearing interferometer, distortions induced into test wave front by optical component aberrations can be constant, so the system error can be erased by calibrations. Since the surface quality of the standard mirror for calibration is 1.6nm, the error induced by optical system will be $e_{ss} < 1.6\text{nm}$.

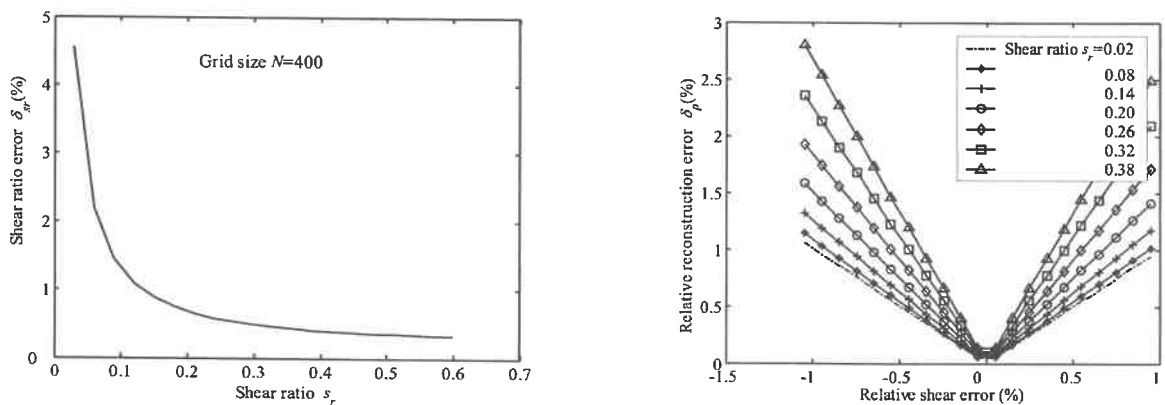
2) Laser stability error in wavelength. Wavelength of laser source is used as the length standard. The adopted Uniphase 1125/p He-Ne laser frequency stability is 300MHz, which means its wavelength drift range is $632.8 \pm 0.00004\text{nm}$. The relative measurement error caused by laser stability therefore can be

$$\delta_\lambda \leq \frac{0.00004}{632.8} \times 100\% = 0.00006\%.$$

3) Shear error. The amounts of shear are obtained by edge discerning of interference patterns. Suppose the two interfering sheared beam size r_b and their relative displacement q in pixels can be obtained by pixel counting, relative error of shear ratio s_r will be [1]

$$\frac{\Delta s_r}{s_r} = \left[\left(\frac{\Delta r_b}{r_b} \right)^2 + \left(\frac{\Delta q}{q} \right)^2 \right]^{1/2}, \quad (1)$$

where the fractional errors Δr_b and Δq are both 0.5. In the adopted LCL-902C CCD, the effective pixel number is 752(H)×582(V), cell size is $8.6\mu\text{m(H)} \times 8.3\mu\text{m(V)}$. The sampling field of the capture card is 576(H)×768(V), and usual sampling image sizes are more than 400(H)×400(V). Fig. 2(a) shows the relationship between shear ratio s_r and shear ratio error Δs_r . By Fig. 2(a) from Eq. (1), when shear ratio $s_r = 0.03 \sim 0.3$ is adapted, the relative error of shear ratio will be 0.5~4%. From Fig. 2(b) the relative measurement error will be 1~4%.



(a) Shear effect on the shear ratio error (b) Shear ratio error effect on the reconstruction result
Fig. 2 Shear effect on measurement result

4) Phase shifting error. In the interferometer, the polarizer P_2 is driven by a stepping motor for phase shifting. Since stepping resolution of the motor is 6400/r, and amplification ratio of the transition gears is 20/100, rotation accuracy of the polarizer P_3 can be $\frac{2\pi}{6400} \times \frac{20}{100} = \pi/16000$, which means the relative phase shifting error is 0.0031%. According to the analysis results by Creath [2], the interferometric phase recovery error by 5-step phase shift algorithm will be ignorable, so is the final measurement error.

5) System noise and sampling digitization error. By experimental testing, the practical noise error level induced to recovered interferometric phase by CCD noise and sampling digitalization is 4.98nm in RMS. According to error propagation characteristics of the improved Zernike polynomial reconstruction algorithms [3] shown in Fig. 3, the induced error will be 1.25nm.

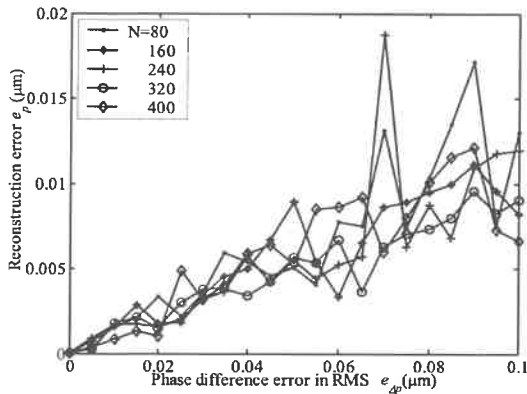


Fig. 3 Effect of noise on the reconstruction

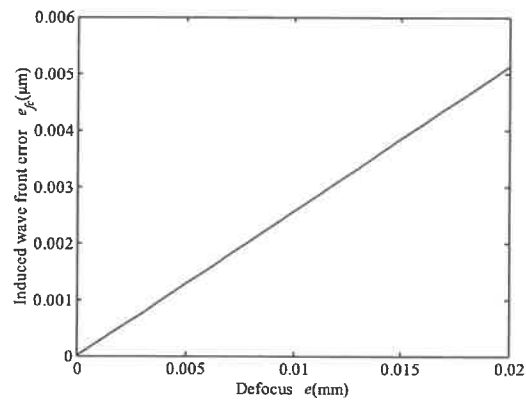


Fig. 4 Effect of the focus coincidence error

6) Effect of focus point coincidence aberration. Focus coincidence deviation of L_1 and L_2 can be limited to $e_v < 0.01\text{mm}$ by manual adjustment in the course of configuration. From Fig. 4, the induced error in RMS will be $e_{fc} < 2.5\text{nm}$ [3].

7) Effect of surface tilt. Error induced by surface tilt $e_{til} < 1 \times 10^{-7}\text{nm}$ [3].

8) Reconstruction error. By improved Zernike polynomial fitting algorithm for circular symmetry wave front reconstruction, relative reconstruction error δ_{re} can be less than 0.002% [3].

The above errors are now summed together to estimate the total error of the interferometer. The errors induced by shear ratio error, laser stability error and reconstruction algorithm are relative contributions: $\delta_{sr} < 4\%$, $\delta_{\lambda} < 0.00006\%$, $\delta_{re} < 0.002\%$. We may sum these errors quadratically and conclude that the total relative inaccuracy is $\delta = 4\%$. The errors induced by optical system, phase shifting error, system noise and sampling digitization error, focus coincidence error, surface tilt and surface position error, are all absolute errors: $e_{ss} < 1.6\text{nm}$, $e_{ps} \approx 0$, $e_{sn} = 1.25\text{nm}$, $e_{fc} = 2.5\text{nm}$, and $e_{til} = 1 \times 10^{-7}\text{nm}$. So for a surface whose surface aberration is $0.2\mu\text{m}$ in RMS is test, the total measurement error will be

$$e_{pl} < \sqrt{(200 \times 4\%)^2 + 1.6^2 + 1.25^2 + 2.5^2 + (10^{-7})^2} = 8.62\text{nm}.$$

Experimental Testing and Discussion

A plane surface whose aberration is $0.074\mu\text{m}$ P-V is measured. Fig. 5 is the obtained bi-directional shearing interference patterns. Fig. 6 is the final measurement result through interference pattern analysis including interferometric phase recovery and reconstruction. Compared the result with the one by a Zygo interferometer, the measurement error is $0.0074\mu\text{m}$. The result of accuracy assessment for the interferometer is confirmed. The repeatability is shown to be less than $\lambda/400$.

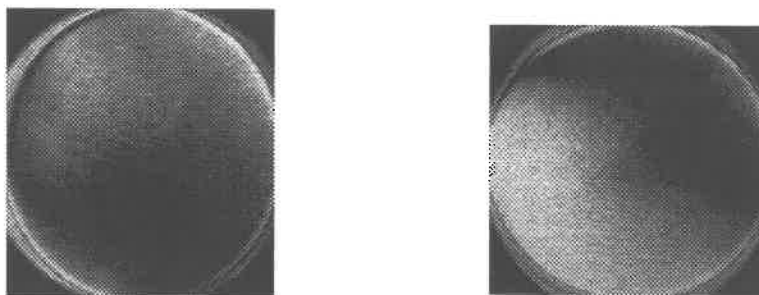


Fig. 5 Bi-directional interferometric patterns

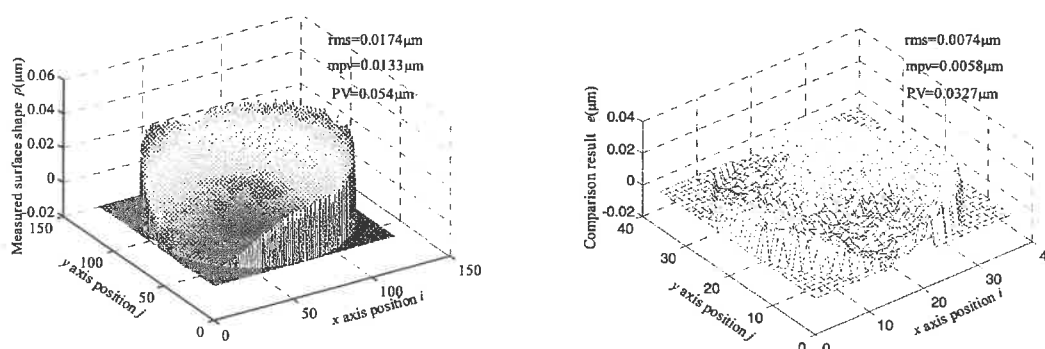


Fig. 6 Comparison of the measurement result with that of a reference interferometer

Conclusions

A new shearing interferometer is proposed for precision surface form error measurement in a manufacturing environment. The principle of the interferometer is introduced and the optical path analyzed. Several error factors in the interferometer are investigated. The results show that the accuracy of the interferometer can reach $\lambda/80$. Experimental testing results on the prototype of the interferometer validated the results of error analysis.

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A SURFACE TEXTURE TOOLBOX FOR TURNING PROCESS DIAGNOSTICS

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1. Introduction

While it is known and accepted that surface texture can be used for process diagnostics, it is rarely done so in the industrial environment. This is primarily because surface finish is not monitored on a continuous basis and surface texture inspection instruments do not have built in tools for predictive applications. Even as sensors and instrumentation for surface texture measurements have improved dramatically over the last ten years, the equipment used in many shop floors has limited features and analysis capabilities. As performance requirements on engineering products are on the rise, there is a need for close monitoring of surface texture to identify any variations in the process that could ultimately affect the function of the component. In this context, this paper discusses a supervised training scheme applied to surface texture parameters for detecting process variations and illustrates their use with an example from hard turning. Surface profiles are collected from a hard turning process while tool state is monitored. Surface profile and tool state information are then used to build a cause-effect model using a supervised training algorithm. The objective in building this model is to be able to predict tool state in future measurements. Testing data is then collected that illustrates the use of the model. Also, the paper presents a prototype version of a turning toolbox that can be incorporated into surface finish inspection equipment on the shop floor.

2. Hard turning application

Surface profiles are measured at these different stages of the tool life during facing in a hard turning process [1]. Two sets of experiments are conducted. The initial set is utilized as a training data set for building a cause-effect model. The second set is utilized as a testing data to demonstrate the applicability of this technique. The state of the tool and cutting force information is also recorded during each run. Table 1 summarizes the cutting conditions. R_a , R_q , PS1 (ratio of second to first harmonic in power spectrum) and PS2 [2] parameters are extracted from the roughness profile after filtering using a Gaussian 80 μ m filter. Because of the availability of binary information on the state of the tool, an attempt is made to relate surface finish parameters with tool state. A variety of mathematical techniques are available to model in such cases. A supervised training scheme has been adopted in this example. An overview of supervised training is presented below followed by results and discussion.

	Speed (ft/min)	Feed (in/rev)	Depth of Cut (in)	Outer Diameter (in)	Inner Diameter (in)
Training	300	0.006	0.008	1.625	1.0625
Testing	450	0.006	0.008	1.625	1.0625

Table 1. Cutting conditions

3. Supervised training of surface finish parameters

Supervised training algorithms [3] can be used in situations where there is limited user knowledge of the underlying physics of the problem, as is often the case in shop floors. It is fairly easy to 'train' an algorithm that learns from similar inputs of known cause-effect relationships. Also, such a training scheme is advantageous in trying to build a generic inference engine that can recognize multiple causes from a single set of inputs. Needless to mention, supervised training requires significant amount of training data that covers the desired range of outputs; this however is not perceived to be a drawback as training data can be gathered during initial setup phase of a new part. The input features extracted from the profiles, R_a , R_q , PS1 and PS2 in this application, are fed to the algorithm. The architecture of the network is dependant on the application. In this case, there are five inputs (four parameters and a bias) and one output neuron. Fig. 1

shows the architecture used in this application. The weights (w_1, w_2) are randomly selected at the beginning. The target values represent the class to which the profile falls. The process of training involves teaching the algorithm to map inputs (R_a, W_a) to its target t over several iterations. The weights are updated according to a certain rule. The process of training is complete if the weights do not differ by a certain predefined tolerance over two consecutive iterations. The bias is also fed as input to the system. The value of bias is always 1 and its weight 'b' is also trained along with the weights of the other inputs. Bias is necessary to shift the origin if necessary. The training rate is set to 0.05 and is used in updating the weights of the system. The training algorithm used is shown in Fig. 2.

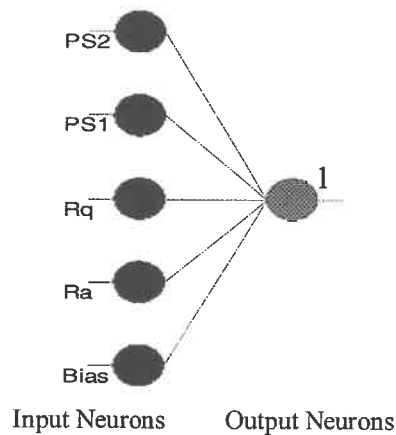


Fig. 1 Architecture used

```

Do until weights converge {
  For input = 1 : n
     $F = 1 * b + R_a * w_1 + W_a * w_2$ 
    If ( $F > 0$ )  $F = 1$ 
    Elseif ( $f = 0$ )  $f = 0$ 
    Else  $f = -1$ 
    If ( $f \neq \text{target}$ ) {
       $W_1 = w_1 + R_a * 0.05 * \text{target}$ 
       $W_2 = w_2 + W_a * 0.05 * \text{target}$ 
       $b = b + 1 * 0.05 * \text{target}$ 
    }
  }
}

```

Fig. 2 Training Algorithm

4. Training and testing results

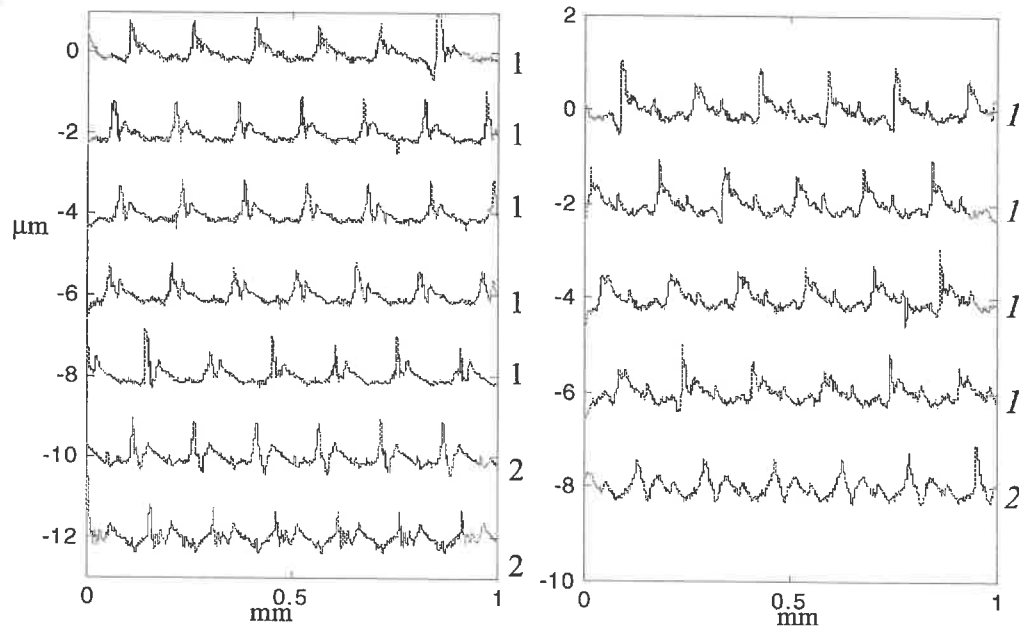


Fig. 3 (a) Training data (from the top: 25, 50, 75, 125, 200, 400 and 600 passes) (b) Testing data (from the top: 25, 50, 75, 125 and 200 passes)

The parameters extracted from the roughness profiles are used to train an algorithm to identify the onset of tool wear. The algorithm is first tested on the training data set to verify the model. Parameters from the

testing data are tested using the algorithm. The algorithm identifies the onset of tool wear at 200 passes in the testing data. Training and testing data are shown in Fig. 3 (a) and (b). The force plots, shown in Fig. 4 (a) and (b), validate the results. As observed from Fig. 4, the increasing trend in the force plots indicates the onset of tool wear and corroborates the results from the algorithm.

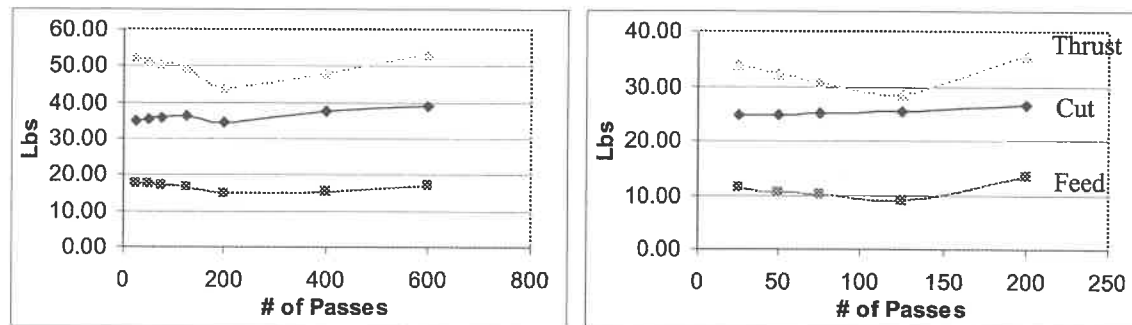


Fig. 4 (a) Force plot in training data set (b) Force plot in testing data set

5. Toolbox

The previous section highlighted an application for process feedback using supervised training routines with surface finish input. This section highlights the use of the toolbox for this application. The toolbox is developed in Matlab® and incorporates the training and testing features described earlier. There are two key user elements to the toolbox –training and testing cycles. The training segment permits the user to upload one profile at a time, input process parameters (tool wear) and submit the contents to the database for later retrieval. After the user has input training data sets, the inference engine processes the data and a model is built between process parameters and metrology data. This completes the training process and launches the testing interface. The testing interface is an extension of any surface finish package commercially available in the market today. The key addition is the recommendation made by the inference engine on the basis of the model built using the training data sets. The details of the toolbox are presented in [4]. Fig. 5 is a snapshot of the training results and Fig. 6 is a snapshot of the testing results. During testing, a sample profile is loaded and the inference engine correctly predicts the occurrence of tool wear during machining.

6. Discussion and conclusions

This paper has discussed the use of supervised training algorithms with surface texture inputs for manufacturing process diagnostics and monitoring. The technique is illustrated with a hard turning application. While the data sets presented here train an algorithm in binary mode, capturing more profile data with relevant process (such as tool wear) information will enable future training for advanced wear, built-up-edge, chatter, axis of rotation and other predictions. The primary objective of the application described in this article is to strengthen the case for a process diagnostics toolbox as a value-adding tool in surface metrology. Close monitoring of surface profiles provides the shop floor engineer with another process feedback tool. As surface finish is a finger print of the process, any variations in the process is reflected in the part; tracking surface texture parameters closely helps in monitoring the process and in diagnosing the source of the variations. Current industrial practice of surface texture is not geared towards predictive applications. A unique feature of this toolbox is its ability in understanding the temporal relationship between several profiles collected over a period of time. Such capabilities are not currently available in commercial instruments that serve as gages for tolerance compliance rather than process diagnostics. There is a need for advanced tools to monitor changes in surface profiles and relate effect to cause and this work is a step in that direction.

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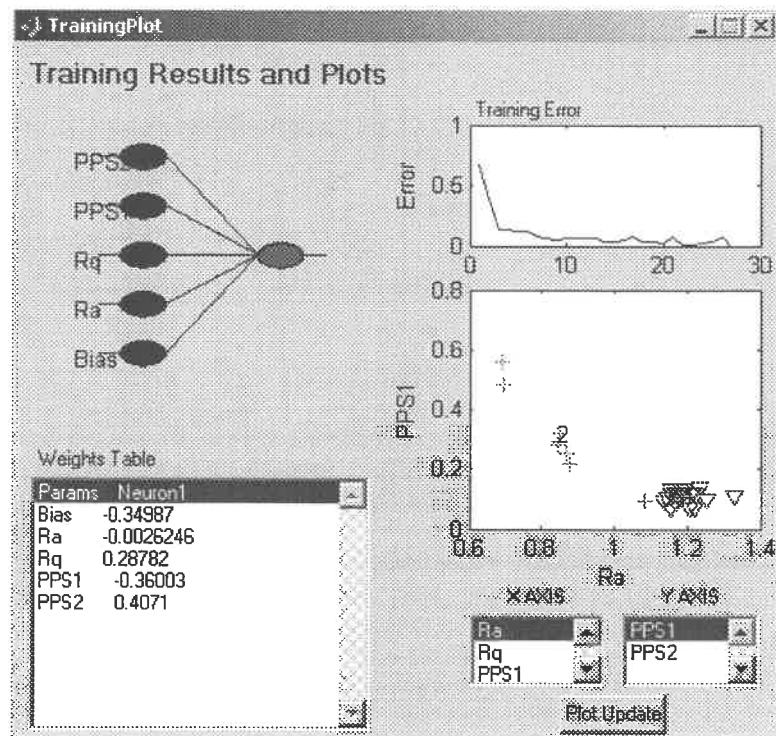


Fig. 5 Training results

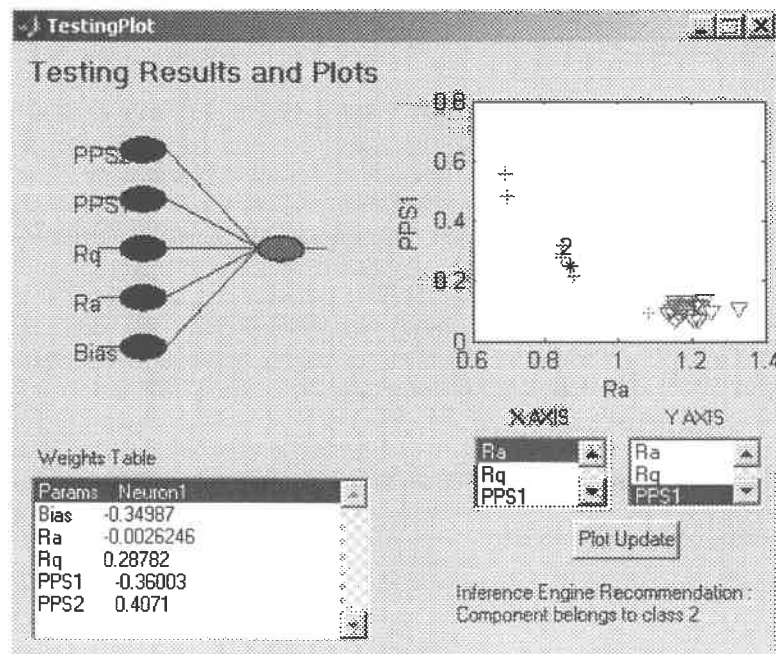


Fig. 6 Testing results

UV Raman Scattering in 6H-SiC Indents

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Introduction

While there have been many studies of the mechanical properties of single crystal silicon, the lack of similar research on single crystal SiC may limit the application of this high temperature, refractory material. In this study, ultraviolet (UV) micro-Raman spectroscopy is employed to investigate the effects of plastic flow and induced stresses in single crystal 6H-SiC subjected to high-pressure indentation. Indentation provides a method to study a material's resistance to plastic deformation under high pressure. The pressures that are achieved at the contact points between the indenter tool and the material can cause the structure of the material's lattice to distort, change form, or fracture. In some cases a small amount of stress, or strain in the lattice, can enhance a device [1]. However, a large amount of stress can directly lead to the formation of voids and cracks in areas extending outward from the point of highest stress. This effect can lead to eventual mechanical failure in devices [2].

UV Raman spectroscopy allows for the nondestructive investigation of materials, and with micro focusing optics the information can be obtained for a small collection volume. Moreover, the short absorption depth of light allows accurate probing of the surface of the indented region. Many other methods of material analysis, which are destructive to the sample, can cause stress relaxation and present misleading results.

Experiment

A Berkovich diamond indenter tip shaped as a three-sided pyramid is used to create triangular shaped indentations. The tip is brought into contact with the material at a constant loading rate of 5mN/sec under a load of 200 mN and 400 mN. Various conditions can affect the resulting plasticity of the material from the indenter. These conditions include the loading and unloading rates, the geometry of the indenter, the indenter size, annealing conditions, and the load amount. All indents were performed on the (0001) surface of the 6H-SiC. This surface corresponds to the Si-face of an ideally terminated structure.

The Raman spectra are collected in a 180° backscattering geometry. The laser power into the microscope is measured as ~50 mW. The laser light is focused with a 40x objective lens to a spot size with a diameter of 600 nm. Visible light is not a good choice of wavelengths for Raman measurements of the surface of this material because it has a photon energy less than that of the bandgap SiC, and the SiC is, therefore, transparent. A UV wavelength of 244 nm is chosen to reduce the laser probing depth to 87.6 nm (where the penetration depth is determined at the point of 10% of the incident intensity), and therefore, reduce the collection volume to $.025 \mu\text{m}^3$. The Raman peaks (center frequency shift and full width at half maximum) are analyzed by fitting to a Lorentzian function or a mix of Lorentzian and Gaussian function for the most accurate fit.

Results

Raman spectroscopy uses monochromatic radiation to detect a difference in energy between incident and transmitted light. The interaction of the electric field component of light with a material can create or annihilate lattice vibrations. The amount of energy absorbed or emitted in a vibration is specific to a material and its bonding structure.

According to quantum mechanics, a vibration is Raman active if the polarizability of the solid changes during vibration [3]. The first-order Raman selection rules for 6H-SiC (uniaxial crystal structure) allow for indirect

observation of lattice vibrations corresponding to the $k \approx 0$ optical phonons. These vibrations include doubly degenerate transverse optical modes (TO), which are derived from atomic displacements perpendicular to the c -axis, and a nondegenerate longitudinal optical mode (LO), which is derived from atomic displacements parallel to the c -axis [4].

In this study, the intense TO frequency at 789.77 Rcm^{-1} (E_2 symmetry) and the LO frequency at 966.79 Rcm^{-1} (A_1 symmetry) are analyzed. (We employ the notation, Rcm^{-1} , to represent the wavenumber shift of the observed mode from the incident laser frequency. In all measurements the Stokes shifted spectra are presented.)

Local stress in a material will typically cause the frequency of the $k \approx 0$ optic phonons to shift to a higher value under compression and a lower value under tension. Figures 1a and 1b present the Raman spectra for the TO and LO modes of vibration of the indented 6H-SiC sample. In comparison to the reference spectrum, which is collected far from the indent, the Raman peaks are shifted to higher frequencies inside and surrounding the indent indicating compressive stress. A maximum shift in the center of the indent signifies that the region of the contact point of the diamond tip and the 6H-SiC is the zone of maximum stress. The stress decreases outward from the center as is indicated by the decrease in frequency shift of the Raman peak.

Discussion

According to prior studies of hexagonal SiC, the mechanical stress can be calculated from the measured shift in the corresponding TO and LO Raman peaks with the following power-law approximation [5],

$$\begin{aligned}\omega_{\text{TO}} (\text{cm}^{-1}) &= 789.77 + 3.11\sigma - 0.009\sigma^2 \\ \omega_{\text{LO}} (\text{cm}^{-1}) &= 966.79 + 3.83\sigma - 0.013\sigma^2,\end{aligned}$$

where 789.77 Rcm^{-1} and 966.79 Rcm^{-1} are the relaxed values in the TO and LO modes respectively and stress, σ , is measured in stress GPa. In the center of the indent obtained with an indenter load of 400 mN, the Raman peak shifts correspond to a stress as high as 2.899 GPa as determined from the TO mode and 1.093 GPa as determined from the LO mode frequency. The stresses calculated from the measured TO and LO optic mode shifts under indenter loads of 200 mN and 400 mN are graphed in Figure 2.

The elastic/plastic contact of a sharp indenter results in a plastic deformation zone of hemispherical symmetry. Tensile stress outside the plastic zone permits outward extension of lateral, median, and radial cracks. Within the plastic zone, compressive stress can terminate crack formation [6]. The detection of compressive stress indicates that the probing depth of the UV Raman measurements is within the plastic zone.

The stress deduced from the LO and TO Raman modes is not equal, however, the difference in the deduced stress becomes closer in the zones of greater stress. The non-uniformity in deduced stress indicates that this material exhibits biaxial compressive stress. Similar experiments that were performed by Liu et al., on 6H-SiC subjected to high hydrostatic pressures in a diamond anvil cell indicate that the splitting reaches a saturation point at ~ 60 GPa. The TO and LO Raman modes were observed to pressures as high as 95 GPa demonstrating the structural stability of this material. It is also reported that the transverse effective charge increases with pressure, reaching a maximum at ~ 40 GPa, and then decreases, suggesting changes in the electronic structure of the material with compression [5].

It is interesting to note that in addition to a peak shift, the peaks show broadening in areas inside and close to the indent. This broadening is attributed to a gradient of stress found within the laser collection volume, and each peak can be fitted to a distribution of Lorentzian peaks. It is possible that a component of the broadening of the TO mode is due to splitting on the double degeneracy of the mode. However, this would require non uniformity in the plane of the substrate, which may be unlikely for the indents. In the stress determinations, the center value of the broadened peak was employed and the possible mode splitting was not considered.

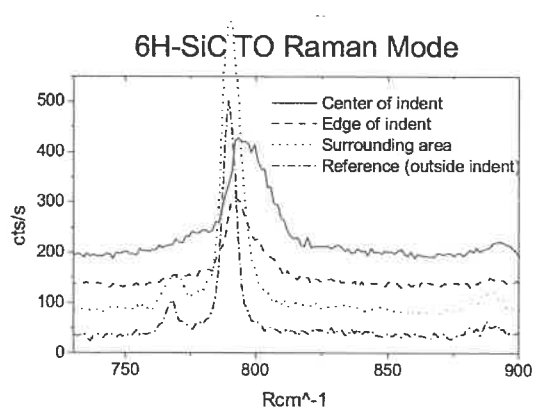
Conclusions

Indentation allows for the study of material plasticity under high pressure. As the demand for products of higher precision is increasing, it is becoming essential to understand the surface deformation of materials on a nanometer scale. Micro-Raman spectroscopy has proven to be a convenient tool for obtaining information about the structure of a material. In this study, the small probing depth of UV light allowed for investigation of the plastic zone caused by indentation. It was found that 6H-SiC subjected to indentation exhibits biaxial compressive stress in regions close to the indentation surface. Stress and stress gradient is greatest near the center of the indent and in the direction perpendicular to the c-axis.

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(a.)



(b.)

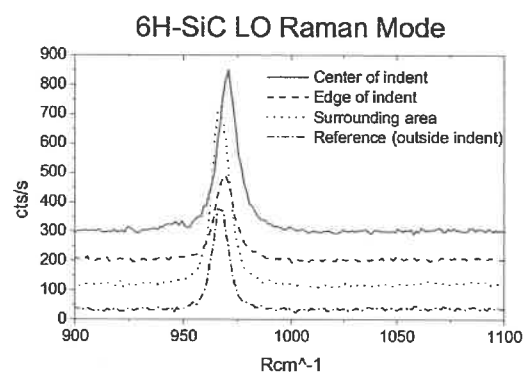


Figure 1: UV micro Raman spectra which display the (a.) TO and (b.) LO modes of vibration for different regions around an indent produced with a load of 200 mN. The reference data is obtained from outside the indent and represents unstressed 6H-SiC. The spectra shifted from this value are collected in regions in and surrounding an indent.

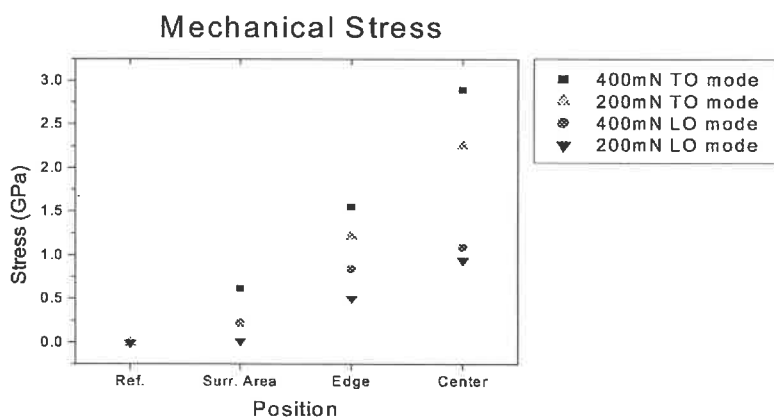


Figure 2: The local stress (in units of pressure) in regions in and surrounding an indent for the transverse (TO) and longitudinal (LO) Raman modes. The results were obtained from indents produced from loads of 200 mN and 400 mN.

Scale-space Analysis of Line Edge Roughness on 193 nm Lithography Test Structures

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1. Introduction

Line edge roughness (LER) is a potential showstopper for the semiconductor industry. As the width of patterned line structures decreases, LER is becoming a non-negligible contributor to resist critical dimension (CD) variation. The International Technology Roadmap for Semiconductors (ITRS) specifies an LER control of 2.4 nm for 2005 and 1.3 nm for 2010[1]. This level of control and the three dimensional nature of LER, pose a challenge to current measurement tools. LER control is a relatively new issue and there are questions on the best way to measure it, on whether the current tools can capture its “real” nature, and more importantly if we need to capture its “real” nature. The use of different roughness evaluation strategies, and the possibility that different measurement conditions with the same instrument may give different values are also causes of concern.

As a part of a study to determine an optimum way to evaluate LER, we measured a series of LER resist test structures made from a single level 193 nm alternating phase-shift reticle containing structures in clear field and dark field tones. The structure includes different periodicity, amplitudes, base linewidths, and sizes of assist features among others. Various types of AFM configurations are being used in order to determine optimum measurement conditions. This experiment requires repeated measurements using different sensor types, tips, scanning configuration and other AFM variables. Here, we present results from our measurements using two different types of tips on a top down scanned tip AFM, and our analysis using scale-space techniques [2].

2. Sample description

A specially designed test structure fabricated by collaborators at International SEMATCH for 193 nm lithography was used [3]. It contains patterned lines of 140 nm and 250 nm linewidth, superimpose with square waves of different frequencies and amplitudes to simulate roughness. The pattern includes both standard binary and attenuated phase-shifted structures. The 250 nm and 140 nm base linewidth square wave features contain both 50 nm amplitude and 100 nm amplitude, and are located at frequencies ranging from 50 nm to 800nm. Figure 1a shows AFM images of an 800 nm spatial wavelength LER pattern. This feature has one of the largest spatial wavelengths on the sample and clearly shows the intentionally inscribed repeating pattern. Figure 1b shows line structures with an intended 250 nm spatial wavelength LER pattern and 100 nm amplitude. In contrast to the image in fig. 1a, the inscribed LER patterns are not clearly visible.

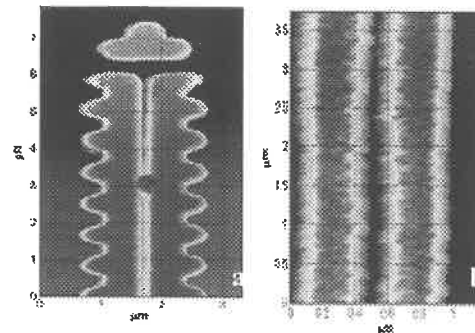


Figure 1: Top view AFM images of line features with inscribed LER patterns. (a) 800 nm spatial wavelength.

3. Measurement conditions

In the first part of the experiment we used two different tips to measure a series of features on our sample. The instrument has a top-down sensor, and uses a scanned tip configuration. The tips used are the so called “conventional-tip” shown in figure 2a, and the so call “sharp tip” shown in figure 2b. It is important to note that these are called “sharp” because of their overall shape. They have the potential to reach cavities where regular tips cannot, and do not necessarily have a smaller radius at the end. Our objective is to see if there are any differences in the line edge profile

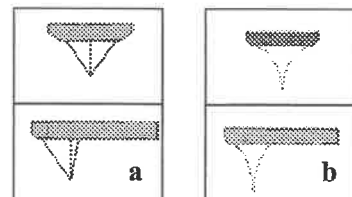


Figure 2: Front and side views of a schematic of a conventional tip (a), and sharp tip (b)

acquired when using these types of tips on a top down scanned tip AFM. Results are presented here for features that were nominally (1) 100 nm amplitude, 100 nm spatial wavelength, 250 nm base linewidth; (2) 100 nm amplitude, 250 nm spatial wavelength, 140 nm base linewidth; and (3) 50 nm amplitude, 50 nm spatial wavelength, 140 nm base linewidth. These represent some of the intermediate and smallest features on the designed pattern, which are most useful for our purposes.

4. Scale-space analysis

Scale-space techniques allow the analysis of structures or signals at multiple scales. The fundamental idea is to derive a group of signals where the high frequency information in each succeeding profile is suppressed. Scale-space operations can be implemented using morphological operators of dilation and erosion. Dilation and erosion operations require two items, an object A (the signal to be analyzed) and a structuring element B. To implement dilation, a new set of points are derived by assigning to each point within the object the maximum value in a region defined by the structuring element. This can be defined as follows

$$(A \oplus B)_x = \max_{\beta \in B} A(x + \beta) \quad 1$$

where the new value of point x (in the original profile) is the maximum in a region defined by the structuring element B when positioned at x. Erosion is also implemented in the same manner; in this case the new point is defined as

$$(A \ominus B)_x = \min_{\beta \in B} A(x + \beta) \quad 2$$

where the new value of point x is the minimum in a region defined by the structuring element B when positioned at x. Using these two operations two filters known as opening and closing can be implemented. To open a signal, the new point is the maximum of the minimum value in the region defined by the structuring element.

$$A \circ B = (A \ominus B) \oplus B \quad 3$$

To close a signal the new point is the minimum of the maximum value in the region defined by the structuring element

$$A \bullet B = (A \oplus B) \ominus B \quad 4$$

The opening filter removes peaks whose widths are smaller than the structuring element, and the closing filter removes valleys whose widths are smaller than the structuring element. Alternating application of opening and closing filters, $(A \circ B) \bullet B$, removes the peaks and valleys smaller than the size of the structuring element. This type of procedure is called alternating sequence filters (ASF), and successively increasing or decreasing the size of the structuring element will result in a group of signals with the finer scales successively suppressed. This produces a multi-scaled view of the data. Later we refer to this succession of signals as levels of the data. Different results can also be obtained by varying the shape of the structuring element used. This type of filtering, which simulates a rolling ball, essentially mimics the scanning of a tip over a surface and may be advantageous in evaluating tip influence. In our analysis we used ball sizes ranging from 40 nm to 640 nm. This represents sizes starting from twice the tip size and larger. Morphological operations are also used in other areas such as image processing [4], and AFM tip reconstruction [5]. Envelope filters, proposed earlier as part of the envelope system [6], of surface assessment are subsets of scale-space analysis. Scale-space techniques are part of an on-going International Standard Organization (ISO) standardization process for use in dimensional metrology [7].

5. Measurements and Results

The results we present here are from measurements at three locations on the sample. Location one is a 100 nm amplitude, 100 nm spatial wavelength LER feature, inscribed on a 250 nm linewidth. The feature at location two is a 100 nm amplitude, 250 nm spatial wavelength feature on a 140 nm linewidth. Location 3 contains a 140 nm

linewidth feature, with an amplitude of 50 nm and spatial wavelength of 50 nm. The image in figure 1-b is from location two.

LER measurement is a relatively new field and what parameters should be measured are still issues of discussion [8]. We evaluated the rms roughness of the LER profile, different levels of scale-space analysis, and the differences of the levels. This highlights some of the differences in the profiles, which were not obvious in the original profile. Figure 3 shows the overall rms values from each of the three locations using both types of tips. Each value represents an average of two measurements in the same location. The rms value for location one was lower for the "sharp" tip (6.46 nm) than for the conventional tip (7.21 nm), however the conventional tip produced lower rms values for locations two and three (7.63 nm versus 6.92 nm and 7.92 nm and 7.23 nm respectively). Each of the sites measured with one tip were relocated, and match locations measured by a different tip type by better than 95 %. However using a decomposition technique (in this case scale-space) one can derive a multi-scaled view of the data that highlights other differences in the signal. The scale-space technique allows the use of a filtering method that simulates the action of the tip; such a method could be advantageous for LER analysis with scanned probe techniques. Figure 4 shows a profile from location 1 (right edge) obtained using a conventional tip (fig. 4(a)) and sharp tip (fig 4(b)). The top profile in each of the plots is the original profile. The rest of the profiles in the plots titled "approximations" are views of the original profiles at different scales labeled with the radius of the circular structuring element. In the "difference profiles" are plotted the differences between the approximation profile at each level and the approximation profile at the preceding level. Figure 5 shows the rms roughness values for the right edge of all three locations using both tips.

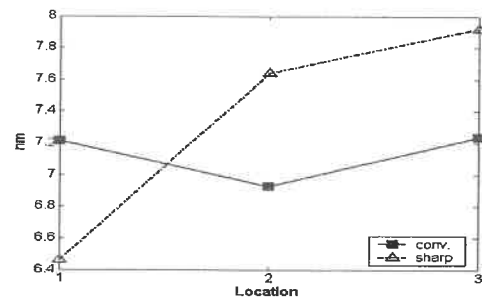


Figure 3: RMS roughness for sharp and conventional tips

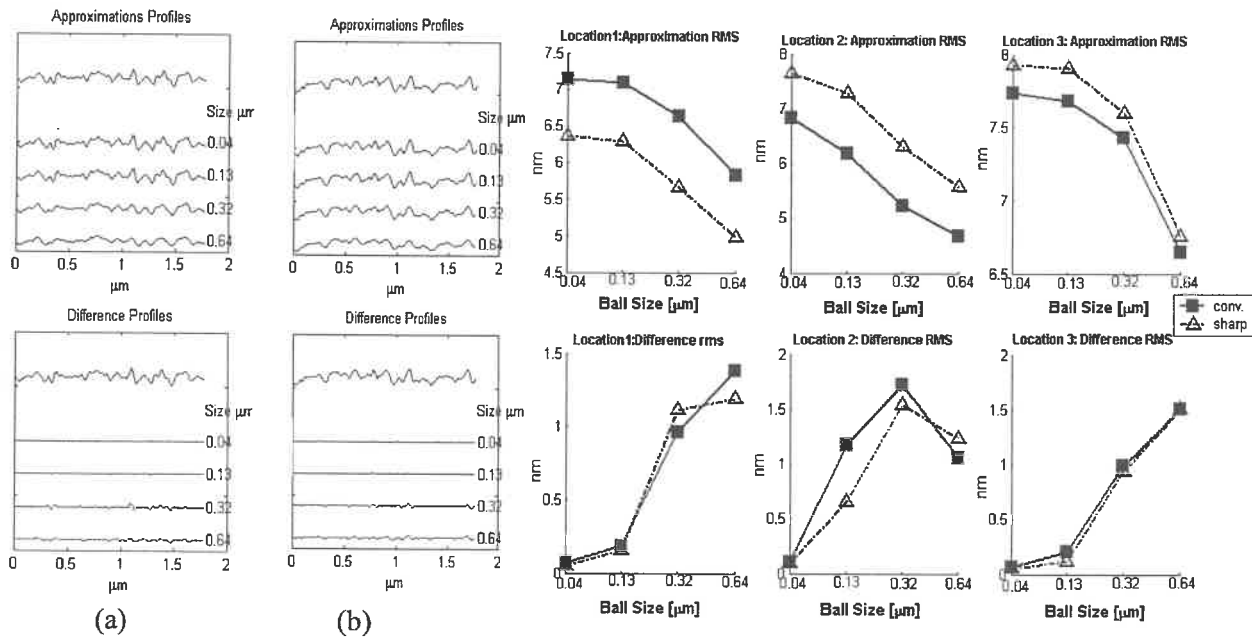


Figure 4 Approximation and difference profiles for (a) conventional and (b) sharp tips from location 2, right edge.

Figure 5: RMS values for approximation, and difference profiles for each location.

In examining the rms values of profiles obtained from the same location using different tips, it is important to point out that we are not directly comparing the original, approximation or difference signals. Rather we are interested in knowing the relative size of the rms value of a profile produced by a specific tip. The rms roughness of the approximation profiles at each location show the same trend as the overall rms roughness shown in figure 3 (for example locations two and three have lower rms values for the conventional tip roughness values in both the

approximation profile and the original profile). The difference profiles allow us to examine differences between two levels of approximations. Generally, the difference profile will depend on the amount of high frequency components present in the signal and the size of the structuring elements used in the approximations. This comes from the way scale-space works. For example, in location one, though the original profile has a higher rms roughness value for the conventional tip, the difference profile at each level will depend on the spatial wavelengths the structuring element is sensitive to. The result is a bandpass filtered profile, in this case produced by a method analogous to the action of the tip itself. Since the signals obtained using conventional tips and sharp tips were processed using the same size and shape of structuring element, this perhaps suggests that the tips are sensitive to different wavelengths of the sample. Other aspects of the imaging process can also contribute to this difference. However under controlled and stable imaging conditions such differences can be mainly attributed to specific parts of the system, in this case the tip. The rms values of the decomposed profiles (figure 5) help quantify the process.

This type of analysis is complementary to spectral techniques like power spectral density (PSD) shown in fig. 6 in both spatial wavelength and frequency domain for the approximation profiles in fig 4(a). Scale-space provides a multi view of the components of the signal. Techniques such as PSD provide time averaging frequency information without indicating the location of the frequency event, and usually only the main harmonics of the signal are clearly represented. For example with Fourier techniques, the location of a small spike in the profile will be suppressed, while scale space will highlight the spike across different scales. The multi-scaled view of scale-space technique provides information about the evolution of the spectrum. This is useful for comparison purposes especially if the surface was produced by a multi-process technique. Scale-space also offers great flexibility in choosing a cutoff relative to Gaussian filters used in traditional surface metrology [9]. For example in figure 5, structuring elements ranging from 0.04 μm to 0.064 μm were used, and we are at liberty to decompose the signal to as many levels as makes sense for our signal and purposes. Another observation from the data in figure 3 is the range of values covered by the conventional tip (6.92 nm to 7.23 nm) which is much smaller than those covered by the sharp tip (6.46 nm to 7.92 nm). Further measurements will determine if this represents a lower bound on the sensitivity of this tip type to this type of sample.

In future work more measurements will be made using other types of sensors, tips and scanning configurations. The overall results will be analyzed to determine the relative differences obtained when evaluating LER using various implementations of the SPM, and also to determine the significance of these differences. The authors are grateful to J. Fu, J. Villarrubia and T. Hu for helpful discussions, and to International SEMATECH for supplying the samples. This project is partially supported by the Office of Microelectronics Programs (OMP) at NIST.

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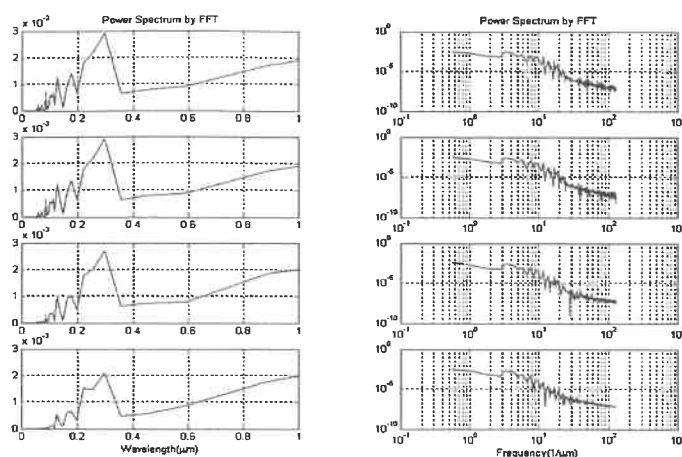


Figure 6: PSD plot for the approximation profiles in fig. 4a (location 1) in both spatial wavelength and frequency domain. There is an apparent periodicity at ~ 290 nm. The peak at $1\mu\text{m}$ corresponds to a longer wavelength on the sample.

FABRICATION OF MICRO GROOVES

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Abstract

Micro-groove is one of the most important features in mechanical, electronic, photonic, and bio-medical components. In recent years, fabrication of micro-grooves became increasingly important with an advancement of product miniaturisation. It is realised that in micro-machining of grooves, the principal problem is the presence of burrs along the feature edges. However, literature survey shows that not much work has been done in the development of cutting grooves process. Thus, in this study, a fly-cut system developed as an attachment on ultra-precision turning machine is used to explore the problems associated with micro-groove machining. Burr formation mechanism is discussed both theoretically and experimentally. With the further development in the process, micro-grooves that are burr-minimized and with mirror finish have been successfully obtained.

Keywords: Micro-machining, grooves, burr formation, fly-cut

1 Introduction

Micro fabrication finds wide application in the field of manufacturing. This is because size and weight reduction can substantially increase the convenience and value of many products. Mechanical micro-machining is the primary choice of various manufacturing processes for fabricating micro components, due to profile accuracy, surface finish, and sub-surface integrity [1-3]. Micro grooves are important features in mechanical, electronic, photonic, and bio-medical components, such as micro lens arrays for optical fiber communication, micro channels for heat exchangers, and micro mesa arrays for bio-medical devices. To fabricate such components, a fly-cut attachment system was developed for the ultra-precision turning machine.

Recently, as deburring has become a bottleneck of the whole manufacturing process, work into burr technology has been activated. From the machining point of view, the formation of burrs and edge fractures during machining means a change of the geometry of parts. Therefore, they are undesirable phenomenon. However, they are also unavoidable. It is classified into roll-over, Poisson, tear and cutoff burr according to the formation mechanism. It is also classified into backward flow, sideways flow, forward flow and leaned burr by the burr formation direction as well as into entrance burr, side burr and exit burr by the location of burr formation [4-6]. Exit burr is one of the primary problems faced during the micro-groove fabrication. Owing to the micro features size, although the burr size is very small, they might cause a tremendous change in the components performance. In addition to burr, cutting tool edge defects adversely affect the accuracy of the groove profile. This paper presents a recent study on the fabrication of micro grooves and minimization of burr formation in micro cutting.

2 Experimental set up

Unlike conventional turning method, where the workpiece is held in the rotary chuck, a fixture was used to hold the fly cutter in the rotary chuck for micro-groove cutting. Figure 1 shows that the workpiece was mounted on a specially fabricated rotary stage, which can also be fed in X and Z directions with a 2.5 nm resolution. This rotary stage enables the workpiece to be mounted at desired angles for the fabrication of special features such as pyramid arrays.

The workpiece materials used in this study were brass, aluminum and copper and all the experiments were carried out with a single crystal diamond cutter.

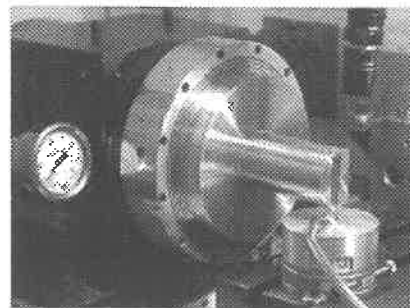


Figure 1 Experimental set-up

3 Burr formation mechanism

The burr formation has been a serious problem in the field of manufacturing [1], especially in micro-machining where the burrs size is comparable to the features size. According to theoretical analyses and experimental observations, the formation of burr can be explained in four stages. In the initial stage, the chip formed in front of the cutting tool is with a positive shear angle. With the advancement of the cutting tool towards the work edge, the plastic zone around the primary shear zone is extended towards the work edge. Since this critical point, the shear angle becomes smaller and a negative shear starts to occur. Here, the distance from the critical point to the work edge is associated with the tool conditions and the work conditions, such as tool edge radius, work edge structure and work material properties. As the negative shear continues and the cutting edge pushes the work material to pivot, the positive shear angle continues to be decreased to near zero degree. With a further advancement of the cutting tool, the work corner will continue to pivot with the chip and finally the chip will be separated from the work material, thus forming burrs.

4 Effects of burr formation and profile accuracy

4.1 Work material effect

There is a serious effect from work material. This is because more ductile materials would not be easily separated from the work edge and may cause bigger burr size. Figure 2 shows the difference of burr size occurring in brass and copper.

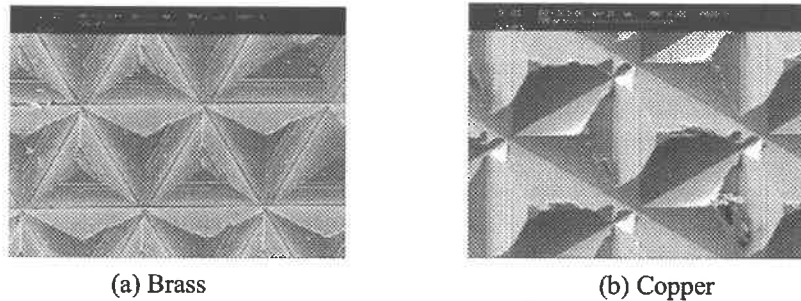


Figure 2 Work material effect

4.2 Tool edge radius effect

A bigger edge radius tool or a worn tool worsens the cutting conditions and cause more severe burr formation. This is because of the reduction in the effective rake angle due to the large tool edge radius caused by edge wear. This leads to smaller shear angle and in turn longer critical length. This will subsequently cause the work material to pivot even earlier from the work edge than that with a sharp tool (see in Figure 3).



(a) Edge radius $r=80$ nm, estimated by AFM (b) Edge radius $r=200$ nm, estimated by AFM

Figure 3 Tool edge radius effect

4.3 Uncut chip thickness effect

The burr height, h_f is related to uncut chip thickness as shown in Figure 4,

$$h_f = (t + l \tan \beta_0) \sin(90^\circ - \beta_0) \quad (1)$$

where, l - length at the starting negative shear; β_0 - initial negative shear angle; h_0 - thickness of the burr [6]. If a proper uncut chip thickness could be chosen, the burr size can be reduced as shown in Figure 5.

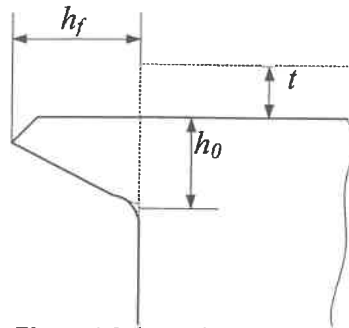


Figure 4 Schematic of burr dimension

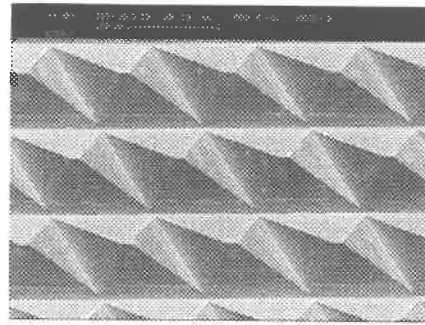


Figure 5 SEM analysis of burrs, uncut chip thickness $t = 5 \mu\text{m}$

4.4 Tool feed effect

Burrs are formed mainly because of the up-cut method employed in micro cutting (see Figure 6 (a)). In this method, the cutting direction is opposite to the feed direction of the workpiece, and at the critical point, the negative plastic deformation start to occur. Thus, some material left at the workpiece edge forms the burrs.

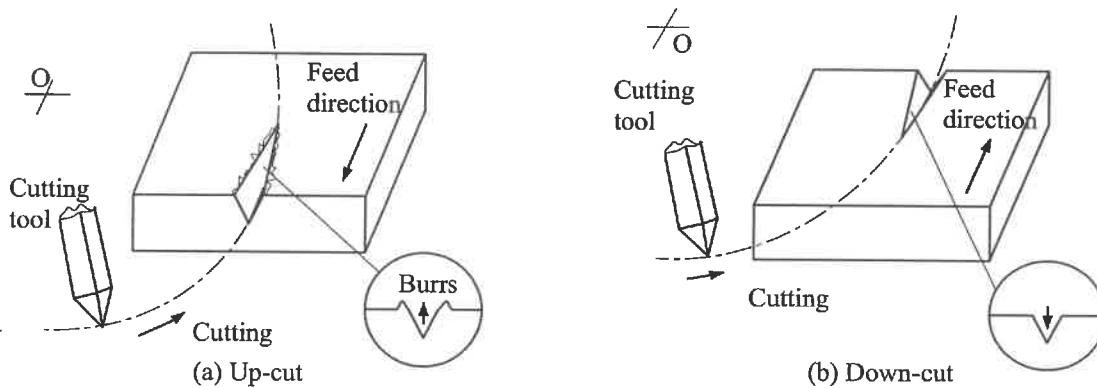
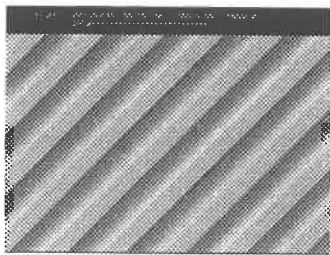


Figure 6 Cutting mode in micro-machining

In the down-cut method, as shown in Figure 6(b), the cutting tool was moved inwards from the workpiece edge. This resulted in comparatively smaller burr size in the machined micro grooves (see Figure 7). This is because in the up-cut method, the undeformed chip thickness increases from zero to maximum value towards the edge of the workpiece, whereas in down-cut, the undeformed chip thickness becomes zero at the tool exit. Also, the critical length is almost zero, thus minimizing the burr size.



Cutting parameters:
Cutting speed: $n = 380 \text{ m/min}$
Feedrate: $f = 60 \text{ mm/min}$

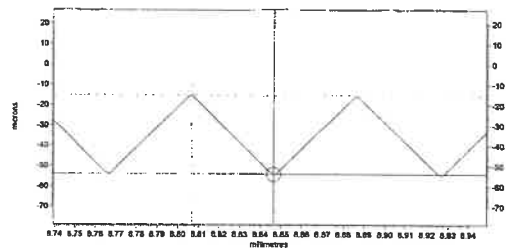
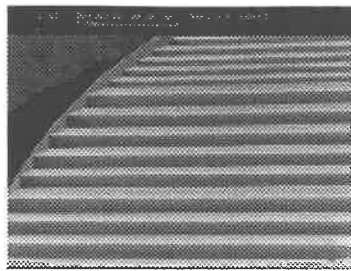
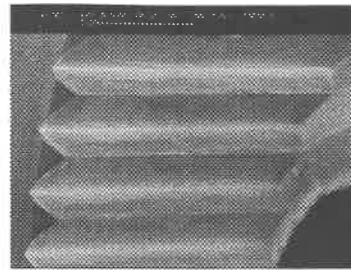


Figure 7 SEM photos and section of micro grooves by down-cut

Some examples of burr-minimized micro machined features using a sharp tool and down-cut feed mode are shown in Figures 8 & 9. Figure 8 shows a standard step height with steps of 0.05 mm, 0.1 mm, 0.2 mm, 0.5 mm and 1.0 mm used for the calibration of instruments. Figure 9 shows a prism array with a pyramid shape, where the pyramid pitch is 80 μm .



(a) Step heights of 0.05-0.2 mm



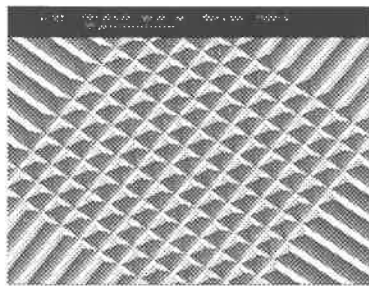
(b) Step heights of 0.5 mm

Cutting parameters:

Cutting speed: $n = 380$ m/min

Feedrate: $f = 100$ mm/min

Figure 8 Standard step heights



Cutting parameters:

Cutting speed: $n = 380$ m/min

Feedrate: $f = 80$ mm/min

Figure 9 Micro prism arrays

5 Conclusions

The primary problem associated with the machining of micro grooves is how to minimize the size of burrs. In an attempt to solve this problem. Effects from cutting parameters, work material, and cutting methodology were studied as follows.

- Work material properties influence the burr size. The more ductile the work material, the larger the burr size.
- With an increase of tool edge radius, the burr size is increased. A sharp tool gives a smallest burr in general.
- A small uncut chip thickness will be beneficial for down sizing the burr formation.
- Down feed should be employed in minimizing burr size.

Further investigation is in progress in various effects and burr formation modeling.

6 Acknowledgements

The authors would like to express their appreciation for the support by A*STAR funded project (U01-P-169A).

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CORRECTIVE FIGURING OF OPTICAL GLASS BY NANO-ABRASION MACHINING

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1. INTRODUCTION

Recently great advances have been accomplished in ultraprecision machining, but it is rather difficult to obtain a few nanometers form accuracy in ultraprecision grinding of ductile materials. To improve the form accuracy, some corrective figuring is performed by local material removal cooperated with feedback of the form error map. The methods used for the local removal are polishing with a small polisher, elastic emission machining (EEM), ion beam machining, plasma assisted chemical machining (PACM), magnetorheological finishing (MRF) and so on.

In previous paper [1], we proposed a new type of local removal method named "Nano-abrasion machining" and investigated its fundamental machining characteristics for different brittle work materials. There it was ascertained that the material removal rate and surface roughness were suitable for a local removal process in corrective figuring.

In this paper, the Nano-abrasion machining is applied to corrective figuring of optical glass and the feasibility is investigated.

2. NANO-ABRASION MACHINING

This method is similar with water jet abrasive machining or liquid honing. In water jet abrasive machining and liquid honing, generally, abrasive grits contained in water are ejected under a high pressure from a nozzle and collide on the surface of work materials at a high speed. Referring to the erosion tests done by Sheldon and Finnie, if the collision energy is low and the collision angle is shallow, the abrasion rate may decrease down to a few nanometers per minute and ductile mode abrasion with a few nanometers surface roughness may occur even for brittle work materials. Therefore this method is named "Nano-abrasion machining".

This method has a great advantage that the machining accuracy is almost independent of the machine accuracy because the working distance from the nozzle tip to the collision point is far enough. A warp of the workpiece induced by holding does not affect the machining accuracy therefore this method is suitable for repeating a post-process measurement and corrective figuring. Additionally, the machining characteristics are stable as there is no tool wear.

3. EXPERIMENTAL APPARATUS AND METHODS

In the machining chamber, the workpiece was mounted on a horizontal X-Y table controlled by a NC controller. The machining liquid was fed to a nozzle by a special screw pump under a high pressure. The work material was optical glass of BK-7. The machining liquid containing 1 wt% abrasive grains of white fused alumina (WA) was used. The grain mesh size and mean diameter were #10000 and $0.6 \mu\text{m}$. The diameter of nozzle was 1mm and the ejecting pressure was 4 MPa, then the velocity of jet flow was 91 m/s. The working distance was 20mm.

After the experiments, the machined surface was observed by an interferometer microscope

to measure the depth profile and surface roughness, then the form i.e. flatness was measured by a laser interferometer.

3.1 Circular motion machining

According to the results of fundamental experiments performed without scanning of the nozzle against the workpiece, when the nozzle was inclined, the removal spot always looked like a crescent moon and the contour map and profiles were not axis-symmetric. The machined area changed its form from oval to circular as the collision angle increased. When the nozzle was set upright, i.e. the collision angle was 90° , the removal spot was circular and the profile had an axis-symmetric W-shape, as shown in Fig. 1(a).

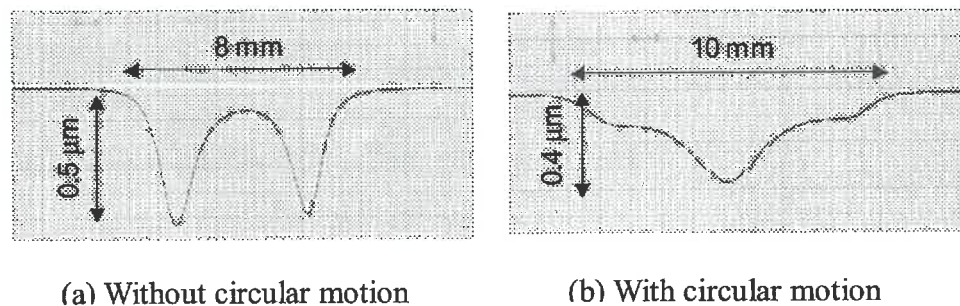


Fig. 1 Diametrical profiles of removal spot without/with circular motion machining
Work: BK-7, abrasives: WA#10000, ejecting pressure: 2MPa, collision angle: 90° , radius of circular motion: 1mm, machining period: 30min.

For a local material removal in corrective figuring, a removal spot with a circular plane figure and axis-symmetric V-shape profiles may be preferable because it is easy to make a simulation program and generate the NC program. To obtain such a removal spot, it is necessary to modify the form of removal spot for collision angle 90° by moving the nozzle or workpiece

In this study, a circular motion of workpiece was employed to realize such a removal spot. To predict the profiles of removal spot, computer simulations were performed for different radii of circular motion, based on the diametrical profile of removal spot obtained for collision angle 90° , as shown in Fig. 1(a).

Then machining was performed with circular motion. Fig. 1(b) shows the experimental result for the best radius, $r = 1$ mm. The obtained profile seems suitable for corrective figuring.

3.2 Simulations of corrective figuring

A computer program has been developed to generate the NC program for corrective figuring and predict the machining accuracy to be obtained.

Fig. 2 illustrates the circular motions and scanning paths in corrective figuring. The circular motion is performed around every machining point distributed with a constant step on the scanning path. To generate the NC program for corrective figuring, it is necessary to determine the velocities of circular motions around every machining point. In this study, the X -direction step and the Y -direction step were 0.54mm equal to the pixel size of flatness measurement by the laser interferometer. The step size a is much smaller than the diameter of removal spot $d = 10$ mm and resultant material removal at a point is a sum of the material removals obtained by machining at the point and a number of neighboring points. As it is very difficult to predict the material removals, the velocities of circular motion at neighboring points were assumed to be

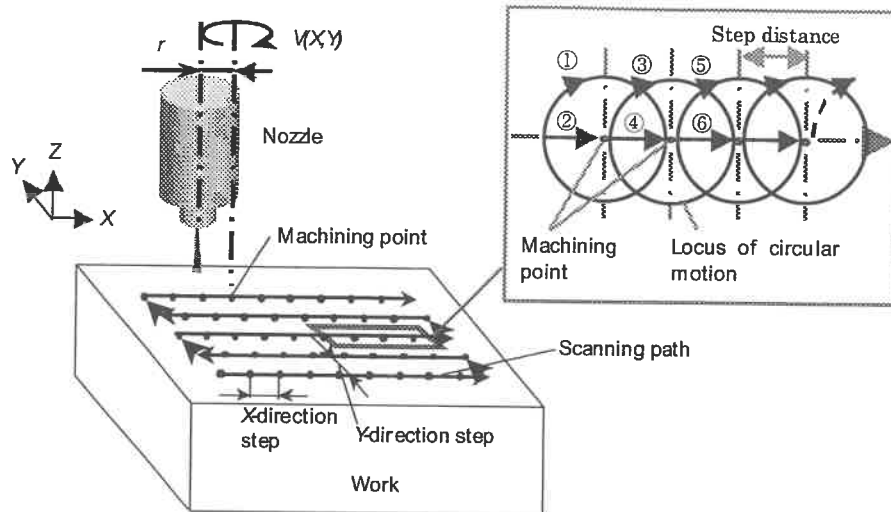


Fig. 2 Circular motions and scanning paths

equal to that at the point. Moreover, the material removal was assumed to be inversely proportional to the velocity of circular motion, following the Preston's law.

In the above calculation, the diametrical profile of removal spot shown in Fig. 1(b) was expressed by two different ways. One was "Cone approximation" where the profile was assumed to be triangular and another was "High order function" where the profile was expressed by a high-order function.

After the generation of NC program, the machining accuracy was predicted and evaluated by computer simulation. Fig. 3 shows an example of predicted machining accuracy. The pre-machined surface had an axis-symmetric sinusoidal convex with a height $\delta_{\max}$ and wavelength λ . The machining intended to make a flat surface. The curve noted "Velocity correction" is the result obtained after correcting the velocities of "High order function" at several points where the residual errors were larger. The larger errors were considered due to the above-mentioned assumption that the velocities of circular motion at neighboring points were equal to that at the point.

From this figure, it is clear that the machining accuracy depends on the ratio of wavelength

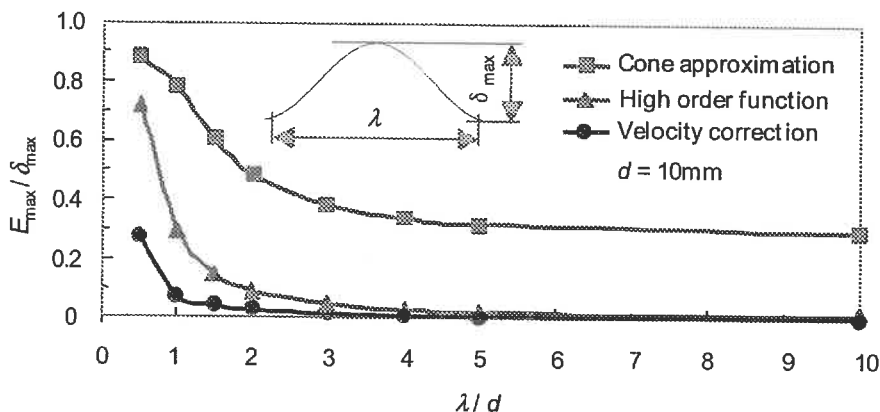


Fig. 3 Machining accuracy predicted by computer simulation

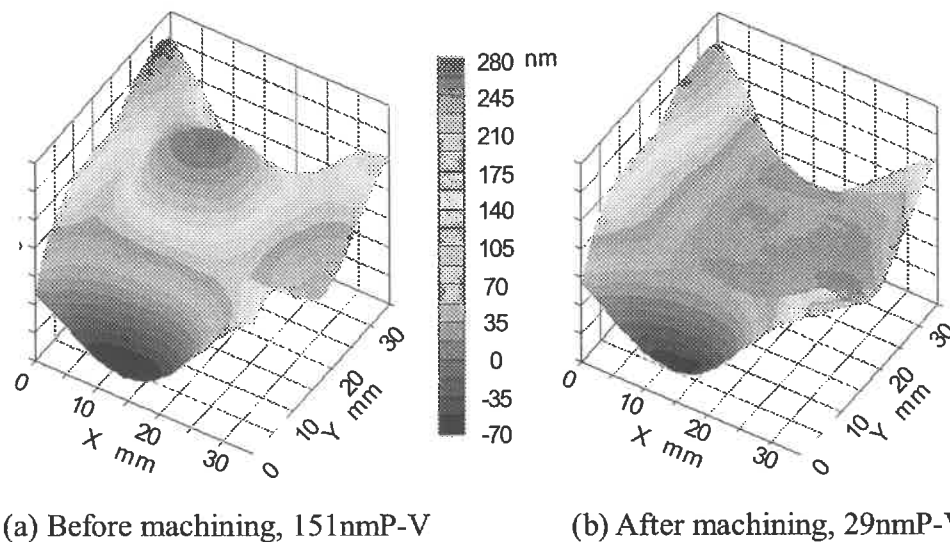


Fig. 4 Experimental result of corrective figuring

of form error λ to the diameter of removal spot d . For example, a sinusoidal form error with 200nm height can be reduced to several nanometers, if $\lambda/d > 1$.

4. EXPERIMENTAL RESULTS

Finally, a series of experiment was performed for corrective figuring. Work material was optical glass BK-7 and the work surface was pre-machined so that it had axis-symmetric sinusoidal concaves with 200nm height and 20mm wavelength. The corrective figuring was performed to make a flat surface of 15x15mm square.

Fig. 4 shows an example of experimental result obtained by corrective figuring of one scanning. The flatness was improved from 151nmP-V to 29nmP-V. The surface roughness was 1.1nmRa slightly better than that of pre-machined surface.

5. CONCLUSIONS

Nano-abrasion machining was applied to corrective figuring of optical glass surface. By modifying the scanning motion, it became easier to generate the NC program for figuring and predict the machining accuracy. It has been ascertained by experiments that the Nano-abrasion machining is applicable to corrective figuring of brittle materials.

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Simulating Vibration Assisted Grinding with Segmented Wheels

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Introduction

Many experiments have demonstrated that vibration assisted grinding can make grinding forces decrease significantly. However, it is difficult to achieve large vibration amplitudes in practice due to the mass of the wheel or workpiece. Therefore, we have to find a way to simulate the large amplitudes. Some researchers have already conducted a series of grinding experiments using segmented wheels. It turns out that the grinding forces also decrease with this special wheel. This paper presents experimental results and theoretical explanations for force reduction with the segmented wheels.

Vibration assisted grinding and segmented wheel grinding are similar in that they both produce intermittent grinding, i.e. sometimes the wheel and workpiece lose contact. Vibration frequency and amplitude can be simulated through the choice of segment spacing and shape. Geometric analysis shows that the finished surface roughness can be made identical for these two grinding methods. Segmented wheels can be made with a relatively deep groove around the perimeter. The segment spacing and shape are infinitely variable, which may prove advantageous when compared to sinusoidal vibration assistance.

Experimental Setup

Three kinds of wheels were used to grind a steel workpiece. Figure 1 depicts the three wheels. The two segmented wheels are cut from regular aluminum oxide wheels. Segmented wheel No. 1 has 20 teeth and segmented wheel No. 2 has 10 teeth.

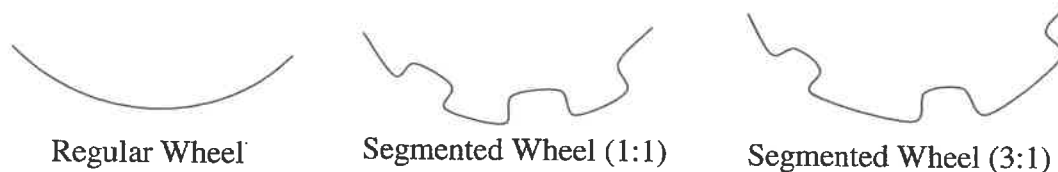


Figure 1: Grinding wheels used in the experiment

Surface grinding was done with a wheel velocity of 29.4m/s. Tests were performed with depth of cut ranging from 5 to 25 microns and feedrates from 33 to 47 mm/s. No coolant was used. Both thrust force F_z and tangential force F_x were measured.

Experimental Results

Figure 2 shows the typical thrust force data for the three kinds of wheels. Note that the forces are lower with the segmented wheels.

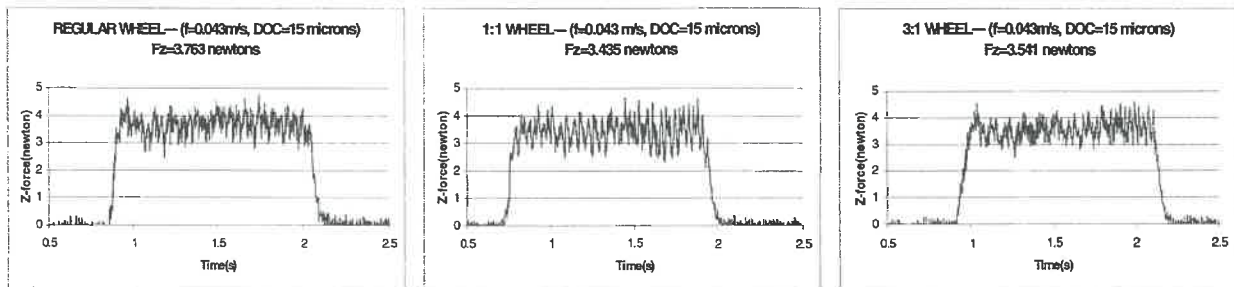


Figure 2: Thrust force results for the three kinds of wheels

Figure 3 shows how the average thrust and tangential forces change with depth of cut and feedrate.

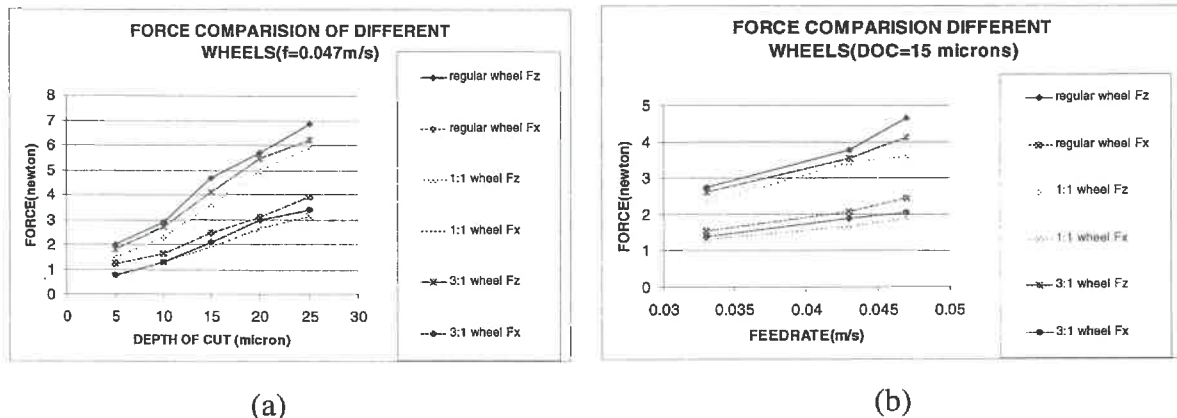


Figure 3: Effect of (a) depth of cut and (b) feedrate on forces

The data in Figure 3 were combined to look at the dependence of force on material removal rate (MRR) as shown in Figure 4. The best fit lines for the three wheels have similar slopes but different intercepts.

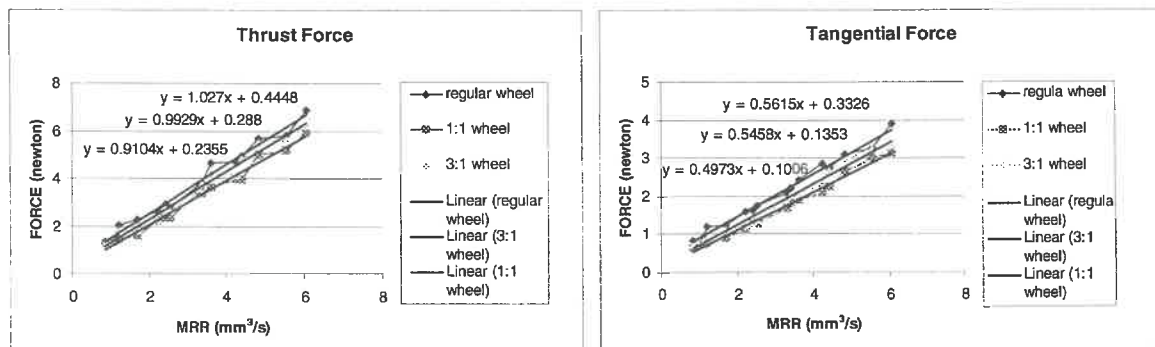


Figure 4: Forces as a function of MRR (Material Removal Rate)

The results suggest that larger segment spacing leads to lower forces. There are several likely reasons. Large segment spacing may facilitate chip evacuation. It may also improve access of the coolant into the cutting zone. In addition, it reduces the amount of time that the rubbing force is active.

Force Model

The grinding force can be modeled as a cutting force, which is proportional to material removal rate (MRR), and a rubbing force, which is independent of MRR:

$$F_z = k_z (MRR) + F_{z0}$$

$$F_x = k_x (MRR) + F_{x0}$$

where k_z and k_x are specific energy constant, and F_{z0} and F_{x0} are the rubbing forces.

Figure 5 compares the material removal rate for the three types of wheels used in our experiment. Using segmented wheels, the workpiece loses contact with the wheel while the wheel groove passes the top of the workpiece. During this period, the grinding force is zero because of loss of contact. During times of contact the forces for the segmented wheels are higher than for the regular wheel.

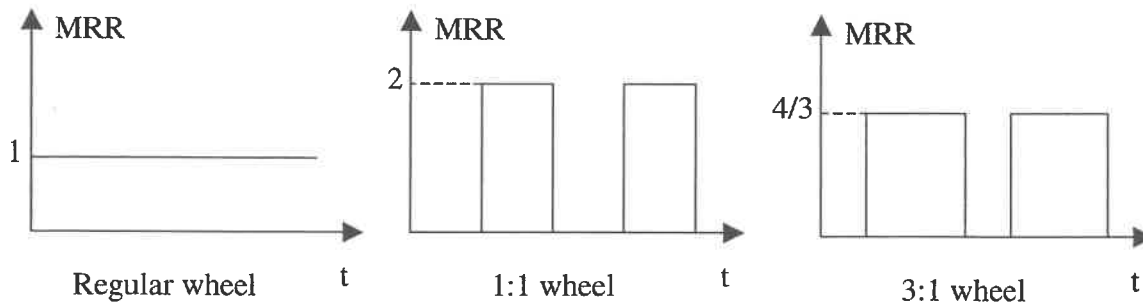


Figure 5: MRR for three kinds of wheels

For the regular wheel, the average grinding forces are:

$$F_{zr} = k_z (MRR) + F_{z0}$$

$$F_{xr} = k_x (MRR) + F_{x0}$$

For the 1:1 wheel, the average grinding forces are:

$$F_{zs1} = [k_z (2MRR) + F_{z0}] / 2 = k_z (MRR) + F_{z0} / 2$$

$$F_{xs1} = [k_x (2MRR) + F_{x0}] / 2 = k_x (MRR) + F_{x0} / 2$$

For the 3:1 wheel, the average grinding forces are:

$$F_{zs2} = [k_z (\frac{4}{3} MRR) + F_{z0}] \times \frac{3}{4} = k_z (MRR) + 3F_{z0} / 4$$

$$F_{xs2} = [k_x (\frac{4}{3} MRR) + F_{x0}] \times \frac{3}{4} = k_x (MRR) + 3F_{x0} / 4$$

According to this model, the slopes of the data in Figure 4 would be constant. This is nearly the case: the thrust force slope ranges from 0.91 to 1.03 and the cutting force slope ranges from 0.50 to 0.56.

The model is not as accurate in its prediction of intercept, i.e. rubbing force. The model predicts intercepts for the segmented wheels that are 0.5 and 0.75 of the regular wheel intercept. Instead the respective ratios are 0.53 and 0.65 for the thrust force and 0.30 and 0.41 for the tangential force.

Suppose k_z and k_x are 0.98 and 0.53, respectively. Based on the model above with $F_{z0}=0.43$ and $F_{x0}=0.24$, the amount of force reduction is predicted. Figure 6 shows the % reduction of the segmented wheels over the regular wheel.

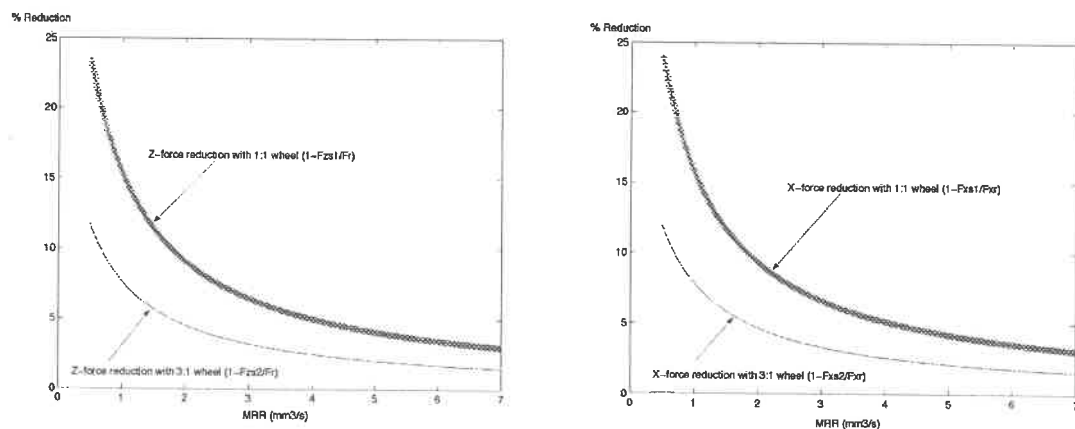


Figure 6: Simulation results of force reduction with 1:1 and 3:1 wheels

Conclusion and Future Work

For all three wheels, the grinding force linearly increases with depth of cut and feedrate. Grinding forces decrease when using segmented wheels due to a net reduction in the rubbing/ploughing component. The model will be refined further to better match the experimental results. In addition, we will investigate wear rates for segmented wheels so that optimal segment geometry can be found.

Investigations Regarding the Interdependence of Spindle Load and Machine Performance During Slicing

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Key Words: slicing, dicing, spindle load, dressing

Abstract: Metal bonded dicing wheels are commonly used for grinding advanced ceramics. But before using such wheels, a preparatory dressing step is necessary to recess the binder and thereby expose the diamond grains. The investigations presented in this paper focused on the wheel's dressing process for high precision grinding of brittle and hard substrates for MEMS applications. A characteristic value for qualifying the dressing process is the spindle load necessary to drive the dicing wheel. By monitoring the spindle load, changes in the dressing process can be observed. Two types of metal bonded wheels, used in combination with two alternative dressing sticks at various dicing parameters were tested and the influence on the spindle load and on wheel's surface conditions are investigated.

1. Introduction

Slicing and profiling by outside diameter high precision grinding are standard technologies for the mechanical micromachining of components for micro-electromechanical systems (MEMS) [1]. An important parameter for the characterization of the slicing process is the tangential force between grinding wheel and workpiece. Measuring this force directly is rather difficult; however for high precision outside diameter grinding ("dicing") machines which are equipped with an air bearing spindle, an indirect approach may be taken: measuring the spindle power and thus the power required to drive the dicing wheel. A variation of the machine parameters or a change of conditions at the dicing wheel led to an alteration of spindle load during the slicing and dressing process. By monitoring the spindle load, changes in the wheel's condition can be detected. The slicing process parameters can be better understood by investigating the interdependence of spindle load and dicing performance [2].

For grinding hard ceramics, metal bonded wheels are commonly used due to the low wear and their potential for high feed rates. For slicing, a preparatory dressing step of the grinding wheel is necessary [3]. During the dressing step, the binder is recessed and thereby the diamond grains are exposed. The purpose of this investigation is to establish the interdependence between the dressing parameters and the spindle load.

2. Experimental Procedure and Results

2.1. Machine and Workpiece Material

For the investigations a commercial dicing machine was used. It is equipped with a precision air bearing spindle designed for accepting dicing wheels with an outer diameter of approx. 2 inches, air bearing guides of the feed axis and a highly rigid machine body. The spindle's rotational velocity is up to 40,000 RPM. A feed rate between 0.01 mm/s and 100 mm/s was chosen. The machine's pitch accuracy, i.e. the accuracy of two parallel dicing cuts, is 0.2 μ m. To minimize the environmental influences, the experiments were carried out in a temperature controlled room at 20 °C (± 1 °C) [4]. For spindle load measurement, a power measurement system was used. To overcome the substantial limitations of commercial available systems, we developed and built our own system and integrated it in the dicing machine. The measurement principle is to simultaneously determine the power on all three phases driving the spindle motor. The material for the dressing experiments was porous SiC. The stick stick material differ as well as grit size, density, porosity, and hardness.

3. Machining Parameters

For the main experiments two types metal bonded wheels and two alternative dressing sticks were used. Three spindle speeds were chosen: 4,000 min⁻¹, 8,000 min⁻¹, and 12,000 min⁻¹. They were matched with three feed rates: 2 mm/s, 5 mm/s, and 10 mm/s. The cut depth into the dressing stick was 1.4 mm and the dressing length, i.e. the cut length in the dressing stick, was 220 mm. To provide stable initial wheel conditions the wheels were trued in a standard process prior to every experiments

4. Results

4.1. Influence of Coolant on Spindle Load

Initial tests were performed to determine the influence of the coolant flow on the spindle load. During these tests, the dicing wheel was not engaged in any cutting process. The coolant flow was 0,7 l/min. Figure 1a shows the spindle load without coolant flow. It shows a non-linear increase of the spindle load with a rising spindle speed. At all spindle speeds, a substantial noise was observed which was strongest at 12,000 min⁻¹. This noise is caused by the dicing machine's inverter. Figure 1b shows a small increase of spindle load due to the coolant flow in the order of 1 to 2 W. The noise also increased slightly in the case of coolant flow. The influence of the coolant flow on the spindle load is negligible and will not be ignored in further examinations.

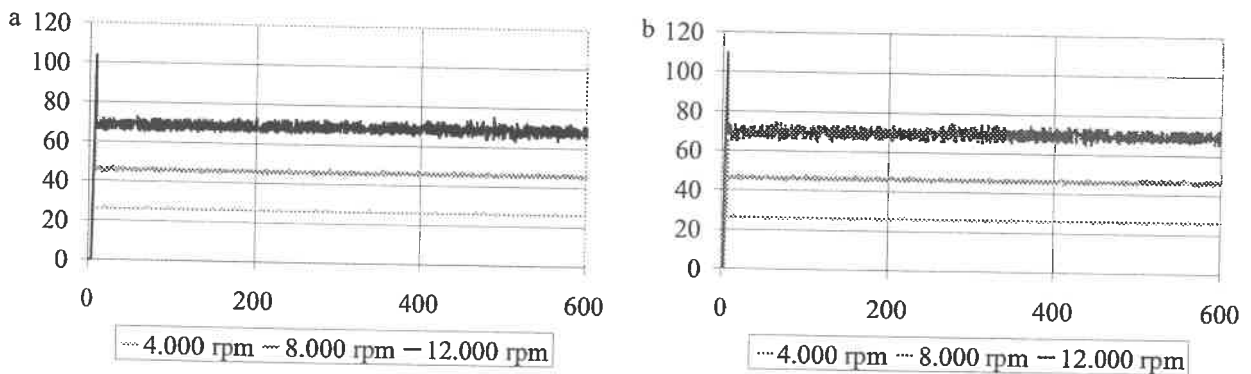


Figure 1: Spindle load without (a) and with (b) coolant flow

4.2. Interdependence between Dressing Parameters and Spindle Load

Experiments were performed to reveal the influence of feed rate and spindle speed as well as dressing stick and dicing wheel on the spindle load. For these tests the dicing wheels cut two times through the dressing stick with a nominal cut depth of 1.4 mm. Thereby, an overall dressing length of 220 mm was achieved for each test.

Figure 2 shows the spindle load signals during two dressing cuts at a feed rate of 2 mm/s. In Figure 2a a dressing stick A was used, Figure 2b depicts the results for dressing stick B. The overshoot at the beginning of each graph is caused by the transition in spindle load at the rated speed. At low spindle speed, the differences between idle load and dressing load is in the range of the signal scatter. At higher spindle speed a visible difference between dressing load and idle load can be observed.

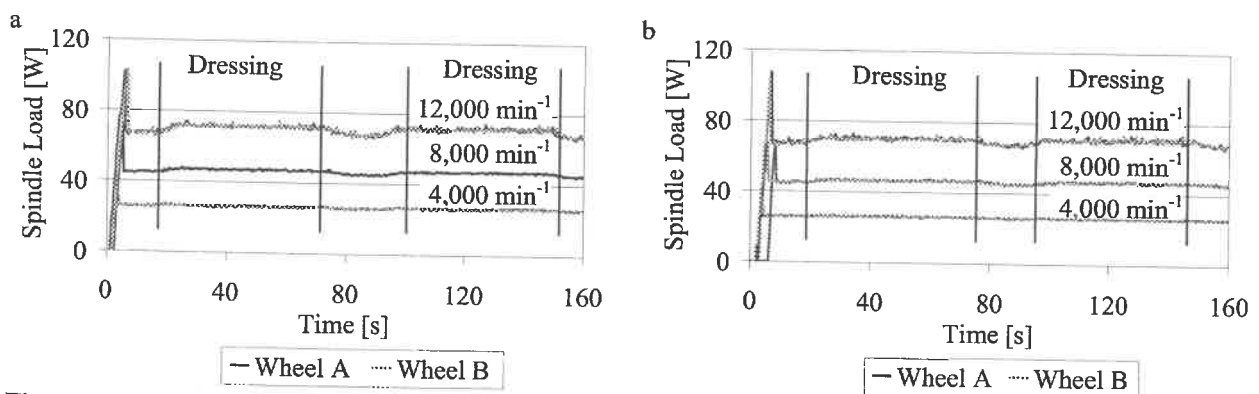


Figure 2: Spindle load signal stick A (left) and stick B (right) at feed rate of 2 mm/s

Figure 3 shows the net spindle load as a function of wheel, dressing stick, and spindle speed. The net spindle load is obtained by calculated the difference between the mean dressing load and mean idle load. A strong influence of the spindle speed on the net spindle load can be found. Contrary to the influence of the wheel, only small differences in net spindle load between wheel A and wheel B was observed. However, stick A caused significantly higher net spindle loads than stick B.

At a feed rate of 5 mm/s the net spindle load is higher than at low feed rate (Fig 3, right), but similar tendencies can be observed. A strong dependency between spindle speed and net spindle load could be found. The noticeably high net spindle load at 8,000 min⁻¹ for the combination wheel B / stick B resulted from a slightly tilted dressing stick. Thus, the dressing depth grew with increasing dressing length which resulted in a rising spindle load.

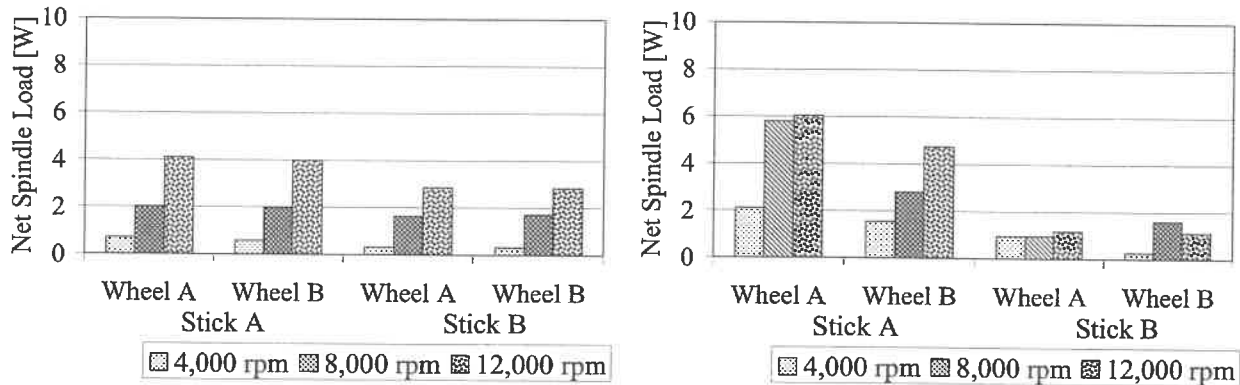


Figure 3: Net Spindle load at a feed rate of 2 mm/s (left), and 5 mm/s (right)

Similar influences on the net spindle load can be found at a feed rate of 10 mm/s (Fig 4). As observed at the other feed rates, the combination of wheel A / stick A causes the highest net spindle load, and the combination wheel B / stick B generates the lowest net spindle loads, independently of the spindle speed.

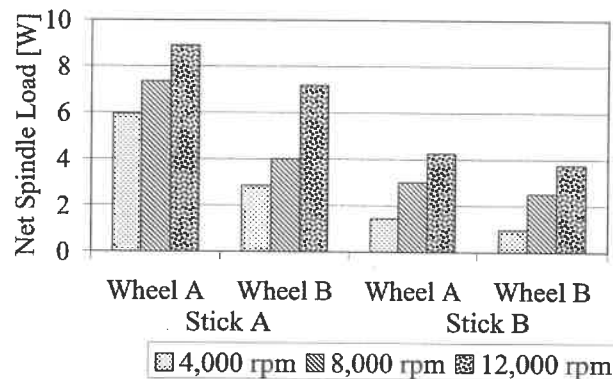


Figure 4: Net Spindle load at a feed rate of 10 mm/s

4.3. Changes in Wheel Conditions

After dressing, the wheels were inspected by scanning electron microscopy (SEM). Figure 5, left reveals the influence of the spindle speed on wheel A's surface condition when dressed with stick B. At a low spindle speed, the diamond grains were completely covered by the binder material. With increasing spindle speed, the binder was more and more recessed and therefore the diamond grains were exposed. As mentioned above, this wheel / stick combination causes the highest net spindle load.

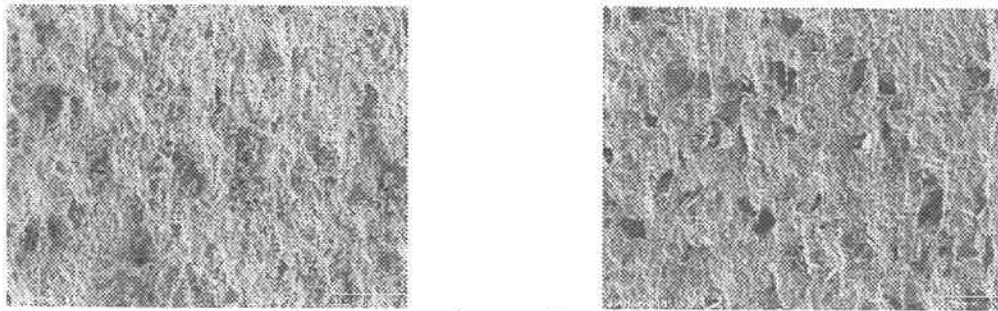


Figure 5: Influence of spindle speed on wheel's surface conditions of wheel A /stick A (left), wheel A /stick B (right)

Investigations of the combination of wheel A and stick B show no influence of the spindle speed on the wheel's surface condition (Figure 5, right). At all investigated spindle speeds the binder is recessed and the diamond grains are exposed.

An similar result could be found with a combination of wheel B and stick A (Figure 6, left). Independently of the spindle speed, the binding material was recessed by the dressing operation, although the grain concentration was lower and the grain size greater than for wheel A. Figure 6, right shows the results after dressing wheel B with stick B. At low spindle speed the binder is not recessed and therefore the diamond grains are not exposed. At high speed, only a single diamond grain protrudes from the binding material in the left picture.

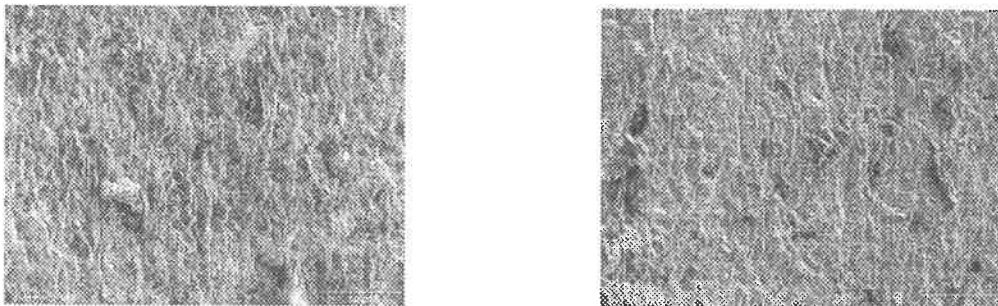


Figure 6: Influence of spindle speed on wheel's surface conditions of wheel B /stick A (left), wheel B /stick B (right)

5. Summary and Conclusions

The executed investigations showed a strong influence of the spindle speed and feed rate on the spindle load. But not only the total spindle load was influenced, the net spindle load (mean dressing load minus mean idle load) depended on these parameters as well. With rising spindle speed and feed rate both total spindle load and net spindles load increased. The combination of dicing wheel and dressing stick greatly influenced the spindle load, too. While one combination caused generally quite low net spindle loads, another one induced significantly higher net spindle loads (up to factor three higher). By subjecting the wheel after dressing to SEM and analyzing the results, a correlation between net spindle load and wheel's surface condition could be found. The wheel / dressing stick combination with the highest spindle loads caused the most recessed binder and thereby the most exposed diamond grains on the dicing wheel while the combination with the lowest spindle load induced the least grain exposure.

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Creep Feed Profile Grinding of High Speed Tool Steels with Ultrafine-Polycrystalline cBN Abrasives

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1. Introduction

Recently, as the high-level grinding systems such as grinding center and ultra-high speed grinding machine and so on are improved, various demands for higher performance of cubic boron nitride (cBN) wheels are increasing [1-2]. In order to enhance the grinding performance of cBN wheels, we have developed a new type of polycrystalline cBN abrasive grit by direct transformation from hexagonal boron nitride, which was produced by chemical vapor deposition process [3]. This new cBN grit not only possesses an ultrafine crystal structure composed of sub-micron sized primary crystal grains, but also has a purity higher than conventional monocrystalline and polycrystalline cBN abrasives [4]. Therefore, it is expected to be used for wider applications than conventional cBN grits.

This paper describes the grinding characteristics of the new ultrafine-crystalline cBN abrasives in the creep feed profile grinding of high-speed tool steels. Experiments for producing a V-shaped groove on a flat surface in one pass by creep feed grinding have been carried out using the new ultrafine-crystalline and representative conventional cBN grits.

2. Properties of Ultrafine-Crystalline cBN Abrasives

Figure 1 (a) shows typical scanning electron microscope (SEM) images of newly developed polycrystalline cBN abrasive grits. This polycrystalline cBN is composed of sub-micron sized primary crystal grains, and its crystal structure is much finer than that of conventional polycrystalline cBN [3-4]. Hereafter, this ultrafine-crystalline cBN is denoted by cBN-U. On the other hand, Figure 1 (b) shows typical SEM images of representative conventional monocrystalline cBN abrasives. This conventional grit is used for comparing with cBN-U grit and hereafter denoted by cBN-B1.

Strength tests on the two types of cBN grit with a grit size of #60/80 were conducted by using the equipment for measuring the fracture strength of a single abrasive grit [4-5]. Figure 2 shows the results of fracture strength tests. Tensile strength of cBN-U is around 3.7 times higher than cBN-B1.

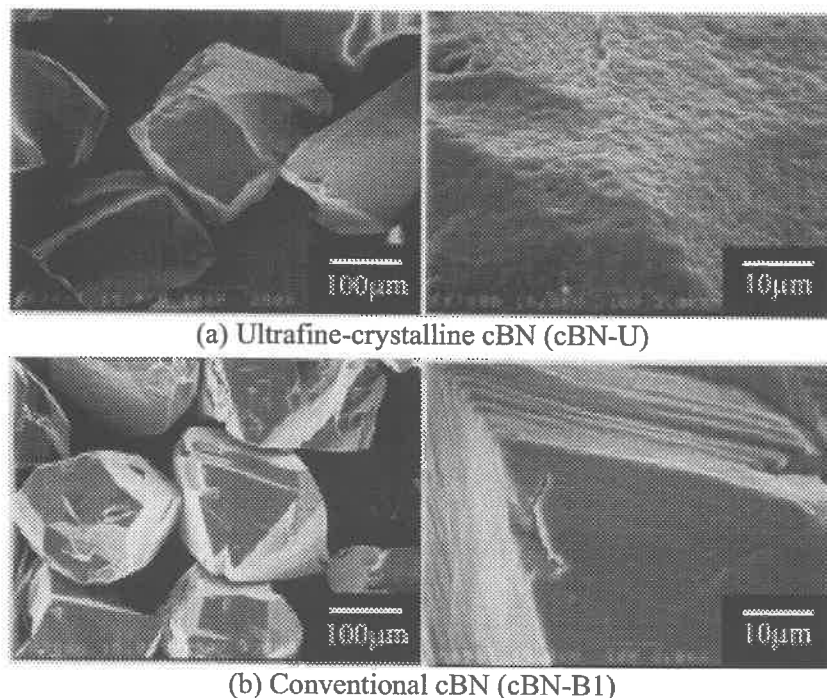


Fig.1 SEM images of ultrafine-crystalline cBN (cBN-U) and conventional cBN (cBN-B1) abrasive grits (Grit size: #60/80).

3. Experimental Procedure

Figure 3 shows a schematic illustration of the experimental setup for the creep feed grinding. In this experiment, a V-shaped groove with a depth of a mm, a V-angle of 60° and a length of l mm is produced on a flat surface in one pass by creep feed grinding. Details of grinding condition are listed in Table 1. Wheel depth of cut and peripheral wheel speed were kept constant at $a = 1.5$ mm and $v_s = 30$ m/s, respectively. For all cBN grinding wheels vitrified bond was selected because of its higher dressability. The grit size was 170/200 US mesh and the concentration was 125. The truing of grinding wheel was performed by using an electroplated diamond block dresser (SD60P), followed by dressing with WA220G stick formed V-shaped groove.

The workpiece material was high speed tool steel (SKH51/JIS). Grinding force components were measured by using a piezo-electric type dynamometer (9257B/Kistler).

4. Results and Discussion

In creep feed profile grinding, it is important to evaluate the profile wear as well as radial wear because it affects the shape and accuracy of ground grooves. The profile wear is

Table 1 Grinding conditions

Grinding method	Creep feed grinding, Down cut
Grinding wheel	cBN170O125VN1 Wheel diameter $d_s = 200$ mm 60° V - face wheel (1EE1)
cBN grit type	Polycrystalline cBN (cBN-U) Monocrystalline cBN (cBN-B1)
Peripheral wheel speed	$v_s = 30$ m/s
Work speed	$v_w = 0.25 \sim 1.5$ mm/s
Wheel depth of cut	$a = 1.5$ mm
Coolant	Soluble-type, 2 % dilution
Groove length	$l = 100$ mm
Workpiece	High speed tool steel SKH51 (HRC65) Dimensions: $100' \times 30'' \times 5^h$ mm

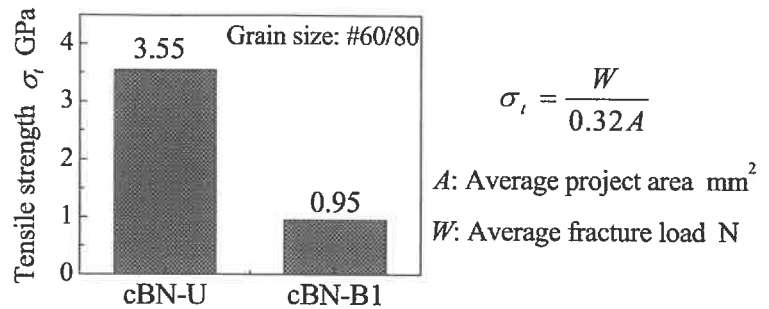


Fig.2 Results of fracture strength tests (Grit size: #60/80).

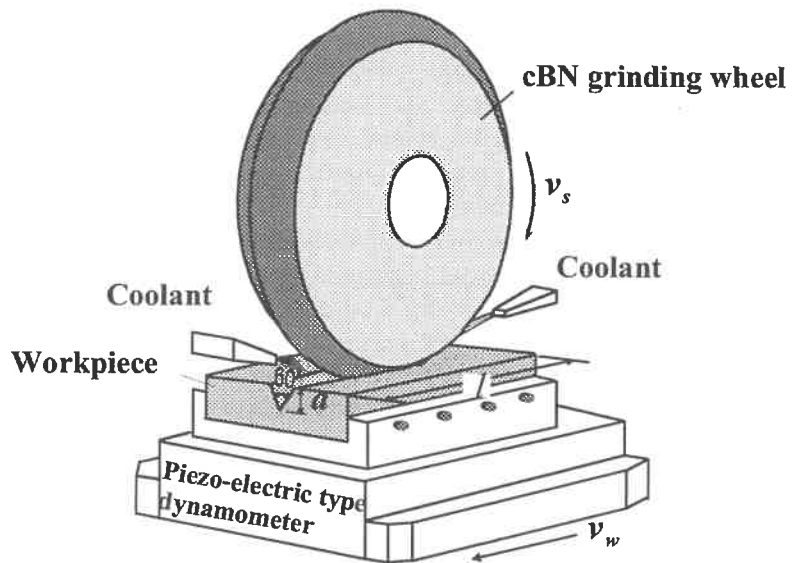


Fig.3 Schematic illustration of the experimental setup for the creep feed profile grinding.

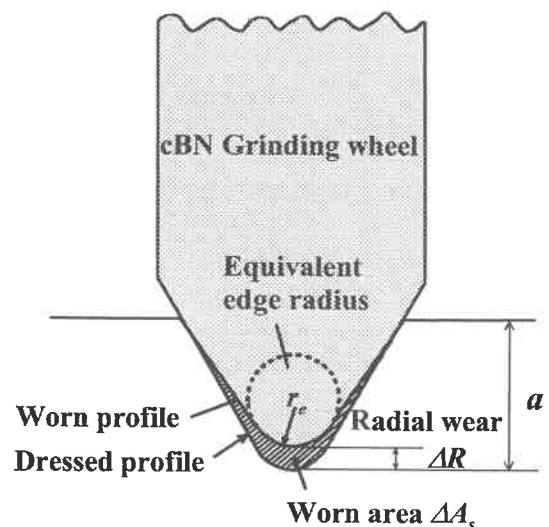


Fig. 4 Illustration of wear analysis around the edge of V face grinding wheel.

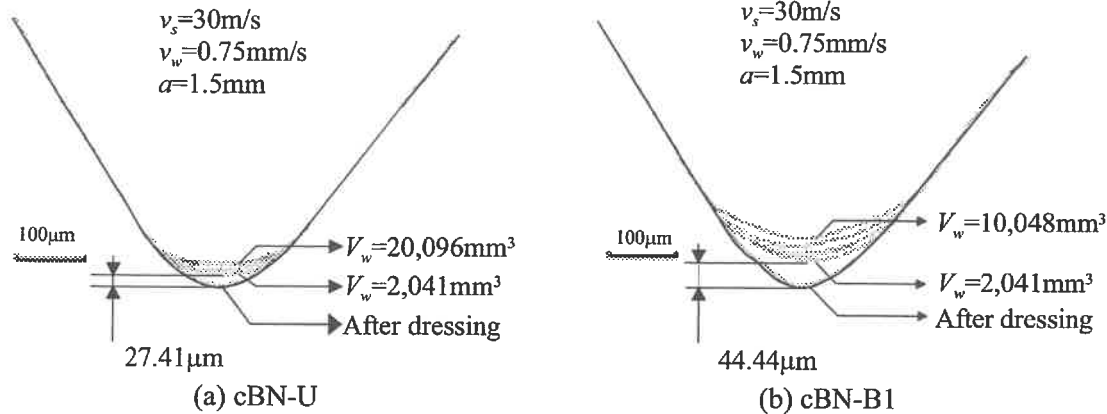


Fig. 5 Changes in the cross-sectional profile of the ground V-shaped groove with the progress of grinding.

the changes of shape of the profile formed on the grinding wheel as a result of wear during the grinding process [6]. The illustration of wear analysis around the edge of V face grinding wheel is shown in **Figure 4**. In this study, the profile wear is evaluated based on equivalent edge radius r_e and volumetric wheel wear V_s . These characteristic parameters for evaluating the profile wear were measured by observing the cross-section of the ground groove using an optical microscope. The corresponding volumetric wheel wear V_s is determined by:

$$V_s = \pi d_s \Delta A_s$$

where d_s is the mean wheel diameter and ΔA_s is the cross-sectional worn area.

Figure 5 (a) and (b) show the changes in the cross-sectional profile of the ground V-shaped groove with the progress of grinding using cBN-B1 and cBN-U abrasives, respectively. In grinding processes with both two types of grinding wheel, the edge radius tends to become larger with the progress of grinding because the wear of edge tip is larger than the wear of both side faces and shows a significant change in the initial stock removal range from 0 to 2040 mm³. However, the change of the edge radius in using cBN-U is very smaller than that in using cBN-B1.

The variations of three characteristic parameters for evaluating the wheel wear which were measured on the basis of these profile forms, with the progress of grinding, are shown in **Figure 6**. When grinding with cBN-B1, the radial wheel wear ΔR and volumetric wheel wear V_s increase rapidly in the initial stock removal range from 0 to 2000 mm³ and then increase gradually at a stock removal range of 2000 to 6000 mm³ (Steady state wear region). But they begin to increase rapidly again after the stock removal exceeds

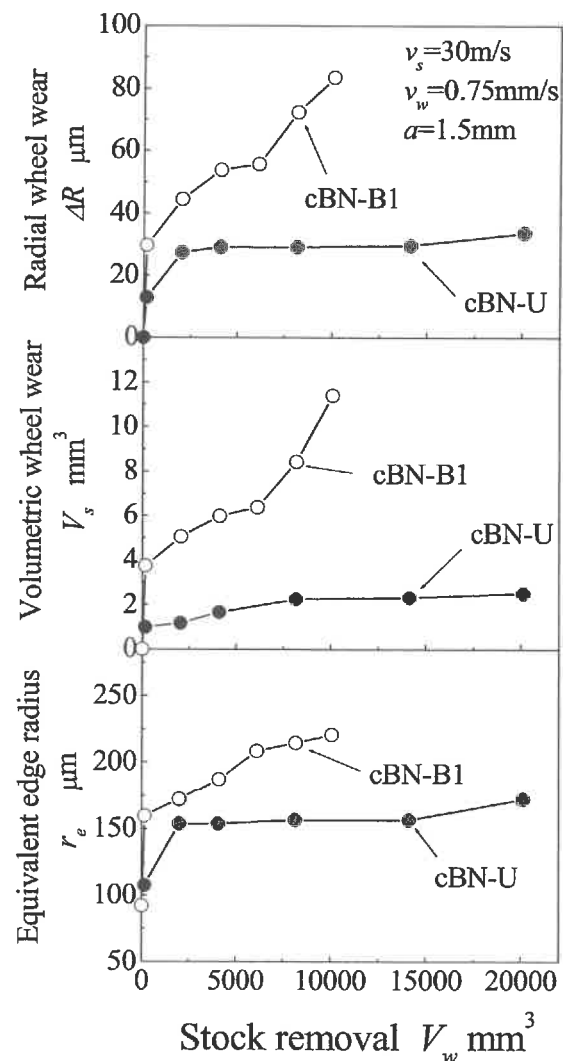


Fig.6 Variations of three characteristic parameters for evaluating the wheel wear.

about 6000 mm³.

On the other hand, when grinding with cBN-U, though the radial wheel wear ΔR and volumetric wheel wear V_s increase rapidly in the initial stock removal range from 0 to 2000 mm³, their values are less than those in grinding with cBN-B1, respectively. Moreover, after the stock removal exceeds about 2000 mm³ the increasing rates of these wear values with an increase of stock removal are significant lower than those in grinding with cBN-B1, respectively.

As the stock removal increases, the equivalent edge radius r_e increases sharply in grinding with cBN-B1, while it retains approximately a constant value after the stock removal of about 2000 mm³ in grinding with cBN-U.

Figure 7 shows a comparison of grinding ratio G_s in grinding processes with cBN-U and cBN-B1 abrasives. Steady grinding ratio G_s was obtained by determining the volume of material removal per unit volume of wheel wear in steady state wear region. Grinding ratio G_s is 3096 in grinding with cBN-B1, while it is 13045 in grinding with cBN-U. Grinding ratio G_s in using cBN-U is approximately 4 times higher than that in using cBN-B1.

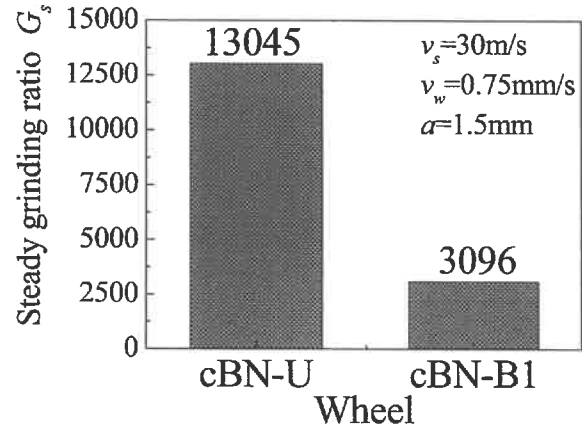


Fig.7 Comparison of grinding ratio G_s in grinding processes with cBN-U and cBN-B1 abrasives.

5. Conclusions

The grinding characteristics of ultrafine-crystalline cBN (cBN-U) abrasive in creep feed profile grinding of high speed tool steels have been investigated and compared with those of a typical conventional cBN (cBN-B1) abrasive. The main results obtained in this study are summarized as follows:

1. When grinding with cBN-U abrasive, both radial wear and profile wear are less, and hence the steady grinding ratio is around 4 times higher than with cBN-B1 abrasive.
2. cBN-U abrasive is suitable for the applications with a high dimensional accuracy in creep feed profile grinding for high speed tool steels, because it gives less profile wear, and hence better form retention, than conventional cBN abrasive.

Acknowledgements

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Enhancement of Grinding Performance Based on Improvement of Uniformity in Structure in Vitrified Bonded cBN Wheels

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1. Introduction

Vitrified bonded cBN wheels have been widely used in various fields ranging from rough grinding to precision grinding because they have higher truing and dressing abilities, compared with non-porous type wheels such as metal bonded cBN wheels. In order to enhance the grinding characteristics of vitrified cBN wheels, many studies on dressing condition, grinding condition [1-2], development of abrasives [3] and so on have been carried out. However, the study of the uniformity in wheel structure based on the manufacturing method of cBN wheels is not yet conducted sufficiently.

The variation in successive cutting-point spacing on the wheel surface decreases by improving the uniformity of wheel structure in vitrified bonded grinding wheels. Therefore, it has been expected that an improvement of the uniformity in wheel structure has enhanced the grinding performance of wheels. However, the relationship between the uniformity in structure and the grinding characteristics is not yet examined sufficiently because it is difficult to make the grinding wheel having such a uniform structure.

In this study, we have proposed a novel manufacturing method of vitrified cBN wheels for improving the uniformity in wheel structure and investigated the influences of the uniformity on the grinding performance of vitrified cBN wheels.

2. New Manufacturing Method of Vitrified Bonded cBN Wheels

Vitrified bonded cBN wheels are generally manufactured in the process as shown in **Figure 1** [4]. First, cBN grains and the liquefied resin are mixed to moisten the surface of cBN grains fully. Next, moistened cBN grains and the vitrified bond powder are mixed to stick the vitrified bond powder on the surface of cBN grains. The mixture (A) is made using such method. After this mixture is passed through the sieve, it is pressed and dried, and sintered at a high temperature. The vitrified bond powder melts and becomes glass, and bond bridges are formed in sintering. The structure of conventional cBN wheels is made in such process.

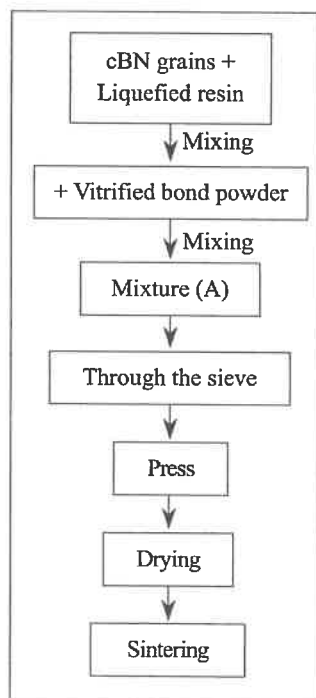


Fig.1 Conventional manufacturing method of cBN wheels (Conventional method)

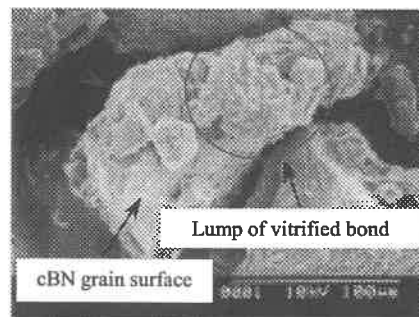


Fig.2 Mixture (A) prepared by conventional method (cBN grain size: #120/140)

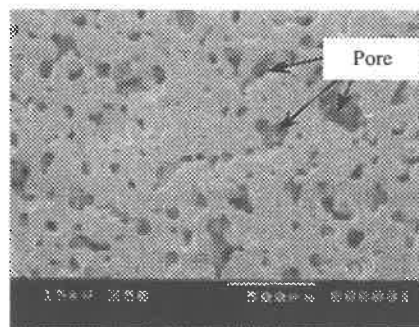


Fig.3 Cross section of cBN wheel made by conventional method (cBN120O200V, $V_p=30\%$)

In order to evaluate the adhesion condition of the vitrified bond powder on cBN grain surface, the mixture (A) made by the method shown in Figure 1 is observed by scanning electron microscope (SEM) (JSM-5310/JEOL). Figure 2 shows a typical example of the SEM image of mixture (A). Used grain is single crystal cBN grain (ABN200/Element six) with a grain size of #120/140. The vitrified bond powder is not covered uniformly on the surface of cBN grains. The surface of cBN grain and large lumps of only vitrified bond powder are frequently observed.

A conventional cBN wheel was manufactured by sintering this mixture (A) using the method shown in Figure 1. This wheel has the percentage of grain $V_g=50\%$ (concentration: 200), the percentage of vitrified bond $V_b=20\%$ and the porosity $V_p=30\%$. A typical SEM observation result of the cross section of the conventional wheel is shown in Figure 3. Flat regions in the Figure 3 are the parts where cBN grains and vitrified bond lumped, and parts of holes are pores. The size of the pore contained in this wheel varies widely. Large flat regions are the parts where lump cBN grains and vitrified bond exist in large amounts. Poor adhesion condition of the vitrified bond powder to the surface of cBN grains is a main cause why the structure of this wheel does not become uniform, as seen from the SEM image of the mixture (A). The part where the surface of cBN grain can be seen in the mixture (A) has a tendency to make a big pore, while the part where only vitrified bond powder lump can be seen in the mixture (A) has a tendency to make the large bond bridge (large flat part in Figure 3). Therefore the wheel manufactured by using the mixture (A) does not form uniform structure.

Considering from the above viewpoint, in order to make the wheel structure more uniform, it is important to cover the vitrified bond powder on the surface of cBN grains uniformly. Therefore we propose a novel manufacturing method of cBN wheel as shown in Figure 4. First, the liquefied resin and the vitrified bond powder are dispersed in the ethanol solvent fully. Next, this slurry is sprayed little by little on cBN grains being stirred. Thus the mixture (B) is made by covering (coating) the vitrified bond powder on the surface of cBN grains. The process after this is the same as the conventional method. A SEM image of the mixture (B) made using this method is shown in Figure 5. All the surfaces of cBN grains are covered uniformly with the vitrified bond powder in this mixture (B). A SEM image of the cross section of the cBN wheel made by using this mixture (B) is shown in Figure 6. Both the size and the distribution of the pore in this wheel are more uniform in comparison with the conventional wheel. Therefore this wheel is more uniform in structure on the whole than the conventional wheel.

Hereafter, the manufacturing method shown in Figure 4 is called "coating method" and wheels made in this method are called "uniform wheels". On the other hand, the manufacturing method shown in Figure 1 is called "conventional method" and wheels made in this method are called "conventional wheels".

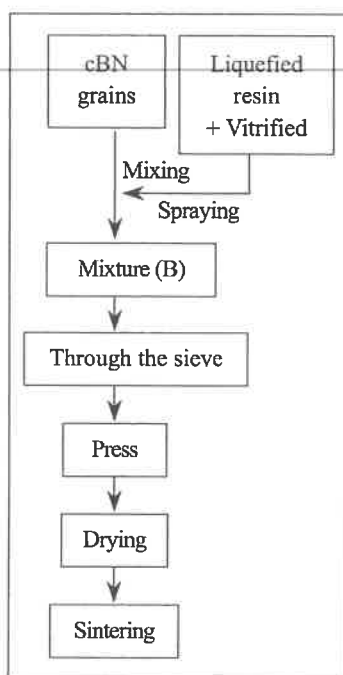


Fig.4 Novel manufacturing method of cBN wheels (Coating method)

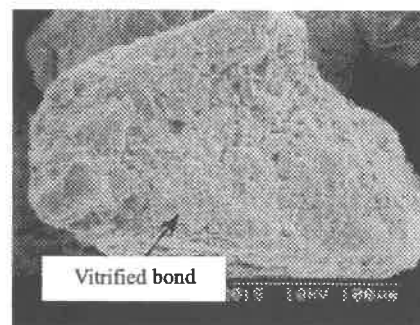


Fig.5 Mixture (B) prepared by coating method (cBN grain size: #120/140)

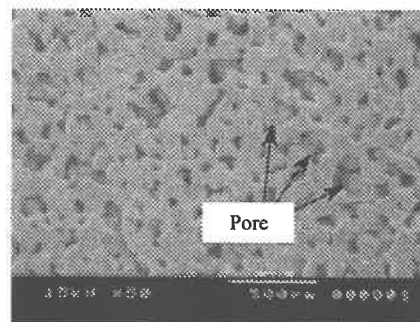


Fig.6 Cross section of cBN wheel made by coating method (cBN120O200V, $V_p=30\%$)

3. Effects of the Uniformity in Wheel Structure on the Grinding Performance

3.1 Experimental Procedure

In order to compare the differences in grinding characteristics between uniform and conventional wheels, cylindrical grinding experiments for high-speed steel steels were carried out using three types of wheel that were made by conventional and novel manufacturing methods, whose compositions are shown in Table 1. A schematic illustration of the experimental setup and the grinding conditions are shown in Figure 7 and Table 2, respectively. The dressing of cBN wheel was performed by using a rotary diamond dresser (Dressing wheel: SD40Q75M) having an AE sensor.

3.2 Results and Discussion

Figure 8 shows the variations of radial wheel wear ΔR with increasing stock removal V_w' in grinding processes with three types of cBN wheel. The effects of wheel type on the relationship between the grinding ratio G_r and stock removal V_w' also are shown in Figure 9.

Radial wheel wear in grinding with uniform wheel-A is smaller than that in grinding with conventional wheel. Consequently, grinding ratio for the uniform wheel-A increases by 15~20 % compared with that for conventional wheel. These results show that the improvement of uniformity in wheel structure is effective to reduce the wheel wear.

Figure 10 shows the effects of the uniformity in wheel structure on the roughness of ground surface R_z with increasing stock removal. The increasing rate of the roughness in grinding process with uniform wheel is lower than that with conventional wheel. The stock removal when the roughness R_z becomes $2.5\mu\text{m}$ is about $10000\text{ mm}^3/\text{mm}$ in grinding with conventional wheel, while it is about $18000\text{ mm}^3/\text{mm}$ in grinding with uniform wheel. Thus, the estimated tool life of uniform wheel, based on the roughness of finished surface, is about 1.8 times longer than that of conventional wheel.

Table 1 Compositions of cBN wheels

CBN wheel	Method of manufacture	$V_g\%$	$V_b\%$	$V_p\%$
Conventional cBN wheel	Conventional method	50	20	30
Uniform cBN wheel -A	Coating method	50	20	30
Uniform cBN wheel -B	Coating method	50	25	25

Table 2 Grinding conditions

Grinding method	Wet cylindrical grinding, Up-cut
Grinding wheel	cBN120O200V Dimensions: $305^D \times 16^R \times 127^H \times 3.5^X$ mm
Peripheral wheel speed	$v_s = 60$ m/s
Workpiece velocity	$v_w = 0.67 \sim 0.52$ m/s (161 rpm)
Plunge velocity (Wheel depth of cut)	$v_p = 8.94$ $\mu\text{m/s}$ ($a = 3.33$ $\mu\text{m/rev}$)
Stock removal rate	$Z' = 2$ $\text{mm}^3/\text{mm}\cdot\text{s}$
Grinding fluid	Soluble type (JIS: W-2-2), 2% dilution
Workpiece	High-speed tool steel (SKH51) Hardness: HRC65 Dimensions: $80^D \times 5^T$ mm

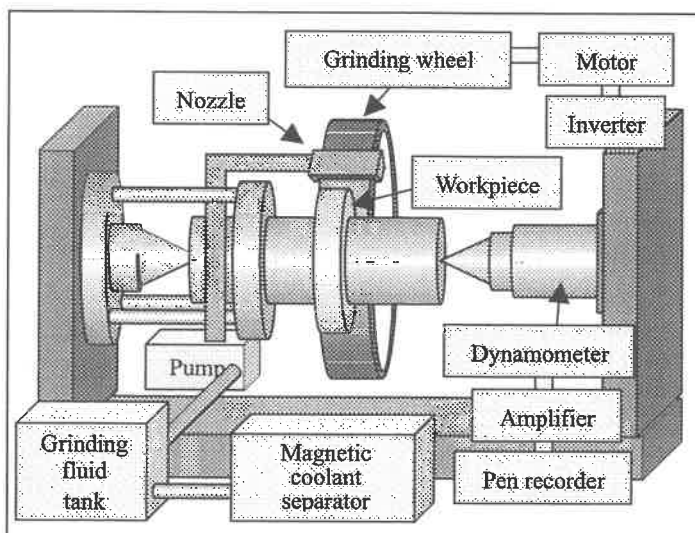


Fig.7 Schematic illustration of the experimental setup

Because the variation in successive cutting-point spacing on the wheel surface decreases by improving the uniformity of wheel structure, the variations in mechanical and thermal loads on the cutting edges in grinding decrease. Therefore the wear of cutting edge takes place uniformly all over the wheel-working surface. Consequently, it seems that the improvement of the uniformity in wheel structure brings on the reductions of the radial wheel wear and the increasing rate of surface roughness with increasing stock removal, as mentioned above.

4. Conclusions

In this study, we have proposed a novel manufacturing method of vitrified cBN grinding wheel for improving the uniformity in wheel structure and investigated the effects of the uniformity on the grinding characteristics of cBN wheels. The main results obtained in this study are summarized as follows:

- (1) The vitrified cBN wheels having an uniformity higher than conventional wheels are produced by the proposed novel manufacturing method.
- (2) The improvement of uniformity in wheel structure is effective to reduce the wheel wear and the increasing rate of surface roughness with the progress of grinding.
- (3) The estimated tool life of uniform wheel, based on the roughness of finished surface, is about 1.8 times longer than that of conventional wheel under the experimental conditions in this study.

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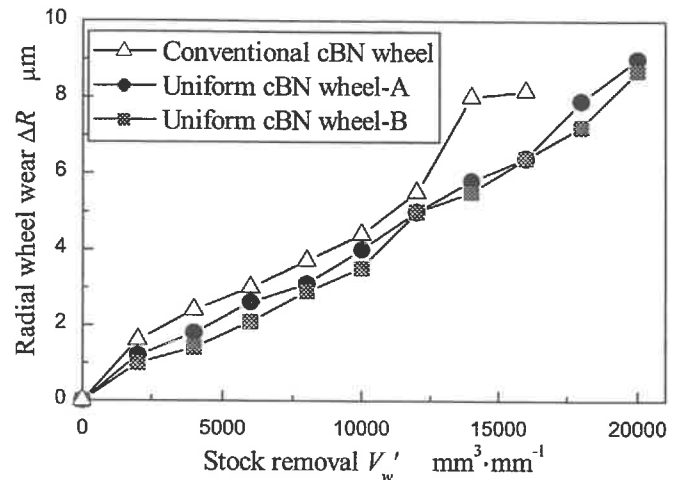


Fig.8 Radial wheel wear ΔR versus stock removal V_w'

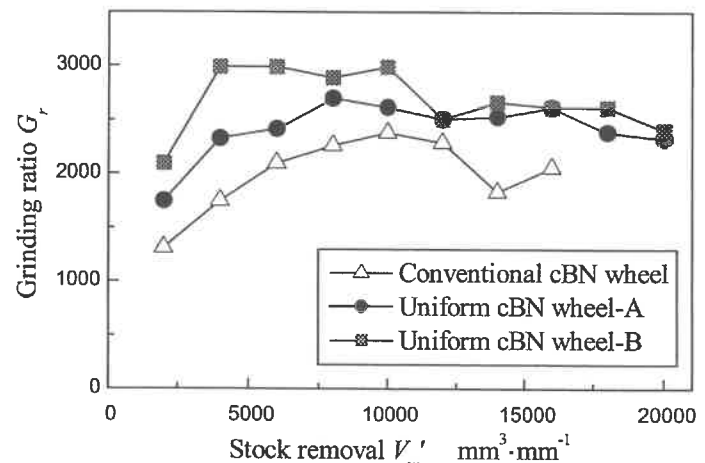


Fig.9 Grinding ratio G_r versus stock removal V_w'

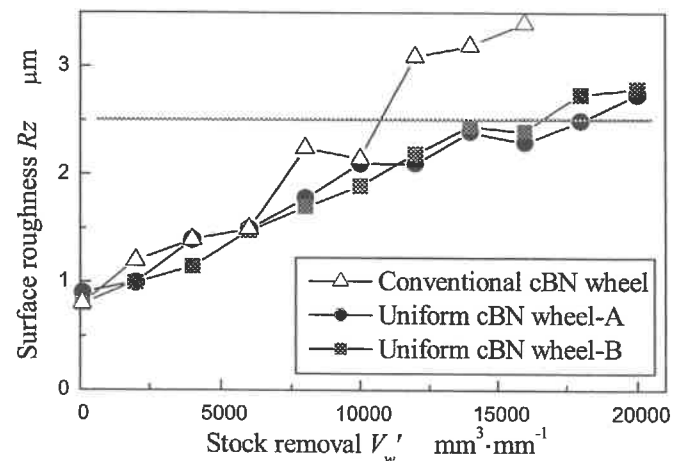


Fig.10 Surface roughness R_z versus stock removal V_w'

Effect of Grinding Fluid Supply on Ultra-smoothness Grinding of Fine Ceramics

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1. Introduction

Grinding operation is one of the most effective manners for high smoothness machining of fine ceramics. However, it is difficult to form crack-free high smoothness surface by ductile-mode grinding because of their mechanical properties of high brittleness. To machine the ceramic component of high quality by low productive cost, the high productive ultra-smoothness grinding technique for the fine ceramics has been strongly required. In our previous research¹⁾, the newly devised ultra-smoothness grinding method is proposed and ascertained to be useful for finishing to near the ultra-smoothness surface. The surface roughness of silicon carbide ceramic and cemented carbide tool formed by the ultra-smoothness grinding method using the #140 diamond wheel is found to attain below 25nm(Rz) or 4nm(Ra), and below 30nm(Rz) or 5nm(Ra), respectively.

This is one of a series of the researches on ultra-smoothness grinding of fine ceramics. In this report, the effect of the grinding fluid supply on ultra-smoothness grinding of silicon carbide ceramic is examined.

2. Ultra-smoothness grinding method

In our previous research¹⁾, the possibility of the high removal rate ultra-smoothness grinding method which can finish to almost the same smoothness formed by polishing is examined using traverse grinding. It is found from the results that the smoothness, which can be attained by traverse grinding, is limited roughly to over the critical surface roughness because of the occurrence of grinding groove parallel to grinding direction formed by grinding. On the basis of the results, the newly devised ultra-smoothness grinding method is proposed as shown in Fig.1 and ascertained to be useful for finishing to near the ultra-smoothness surface. First of all, in the new method, the workpiece-wheel contact width is ground by feeding the workpiece toward the direction normal to grinding direction. As shown in Fig.2, the feed per a wheel revolution, f_{cn} , is smaller than the wear width of cutting edge, w_n , normal to grinding direction. In the second step, after the workpiece is slightly step-fed to the length of f_p parallel to grinding direction so that the geometrical surface roughness formed by overlapping the cross-section of two wheel circles before and after the step-feed becomes smoother than the required surface roughness, the workpiece-wheel contact width is ground again by feeding reversely the

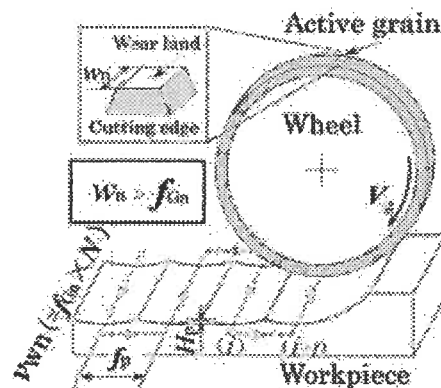


Fig.1 Ultra-smoothness grinding method

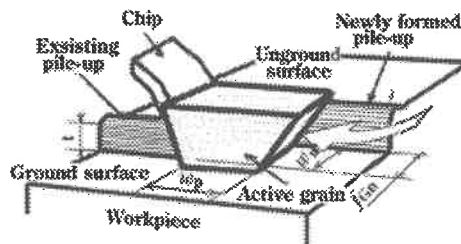


Fig.2 Mechanism of ultra-smoothness grinding

the geometrical surface roughness formed by overlapping the cross-section of two wheel circles before and after the step-feed becomes smoother than the required surface roughness, the workpiece-wheel contact width is ground again by feeding reversely the

workpiece toward the direction normal to grinding direction. The whole surface of workpiece is finished by repeating such a grinding procedure.

3. Experiments

The experiments are carried out with the NC grinder as shown photographically in Fig.3. The grinder using air spindle has the accuracy of $0.1\mu\text{m}$ for each movement of the X, Y and Z direction. The experimental conditions are summarized in Table 1. The grain size and concentration of the wheel used are #140 and 50, respectively. The hot pressed silicon carbide (HPSC) is used as the workpiece of fine ceramic. All of grinding fluids are soluble type. Type T1 includes a plenty of anionic surfactant. Type T2 includes a plenty of fatty acid and a little ester and glycol. Type T3 includes a plenty of glycerin. The observation and roughness measurement of the ground workpiece surface are done with Nomarski microscope and SEM, and with the surface interferometer (WYKO TOPO-3D), respectively.

4. Results and discussions

4.1 Effect of grinding fluid supply

Figure 4 shows the microscope and SEM photographs of the HPSC surfaces formed by ultra-smoothness grinding in dry grinding and wet grinding. The grinding condition is shown in the Figure. From the figure, it can be confirmed that the grinding grooves observed in traverse grinding can't be observed on both of dry and wet ground surfaces. In dry grinding, however, the surface

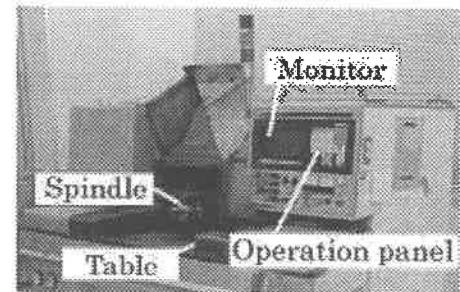
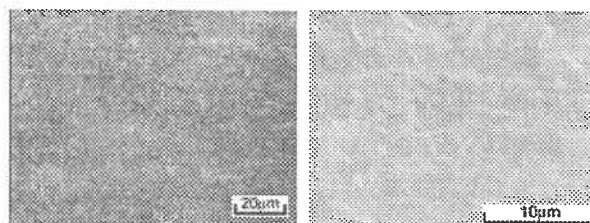


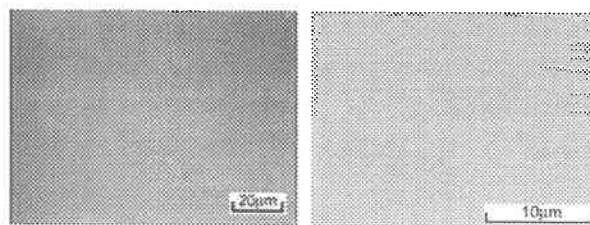
Fig.3 Appearance of NC grinder

Table1 Experimental conditions

Work piece	HPSC
Wheel	SD140Q50M
Wheel speed : V_g	20m/s
Vertical feed : f_{Gn}	10mm/rev
Table speed : v_{wn}	0.63mm/s
Parallel feed : f_p	10 μm , 30 μm
Depth of cut : t	5 μm
Grinding fluid	Soluble type(T1,T2,T3)
Dilution degree	1/10, 1/30, 1/50
Flow rate	12 l/min



(a) Dry Grinding



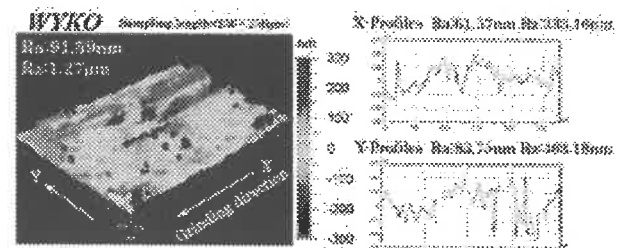
(b) Wet Grinding

Microscope ← SEM

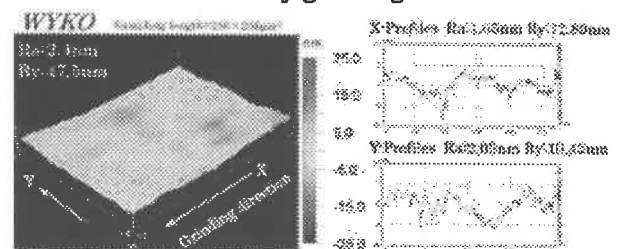
Grinding direction

Fig.4 Comparison of the surface ground of HPSC formed by dry grinding with wet grinding

[Ultra-smoothness grinding $V_g=20\text{m/s}$, $f_{Gn}=10\mu\text{m/rev}$ ($v_{wn}=0.63\text{mm/s}$), $f_p=30\mu\text{m}$, $t=5\mu\text{m}$]



[Dry grinding]



[Wet grinding]

Fig.5 Comparison of Surface roughness of HPSC ceramics surface ground formed by dry grinding with wet grinding

[Ultra-smoothness grinding $V_g=20\text{m/s}$, $f_{Gn}=10\mu\text{m/rev}$ ($v_{wn}=0.63\text{mm/s}$), $f_p=30\mu\text{m}$, $t=5\mu\text{m}$]

topography like melted surface layer is observed on the workpiece surface. In the wet grinding, on the other hand, the smooth ductile-mode ground surface without grinding cracks is obtained all over the whole workpiece.

Figure 5 shows WYKO 3D images and 2D profiles of 256 μ m square surface of HPSC ceramics formed by dry and wet ultra-smoothness grinding. In case of the dry grinding, the 3D surface roughness is about 1.3 μ m(Rz) or 92nm(Ra). From the 2D profiles, the 2D surface roughness parallel to grinding direction is about 333nm(Rz) or 61nm(Ra). And also the 2D surface roughness normal to grinding direction is about 463nm(Rz) or 84nm(Ra). In case of the wet grinding, on the other hand, the 3D surface roughness is about 47nm(Rz) or 2.4nm(Ra). Both of the 2D surface roughness parallel and normal to grinding direction are about 10nm(Rz) or 2.0nm(Ra). The smoothness in wet ultra-smoothness grinding is found much better than dry grinding. It is considered from these results that grinding fluid supply has large influence on the workpiece surface formed by ductile-mode ultra-smoothness grinding.

In Figure 6, the Vickers hardness of the workpiece surface layer formed by dry grinding and wet grinding is compared. For the comparison, the Vickers hardness by polishing is also shown in the figure. From the results, the hardness formed by both of dry grinding and wet grinding is found lower than polishing. The surface layer in dry grinding is softer than wet grinding. It is considered from the result that grinding heat softens the surface layer.

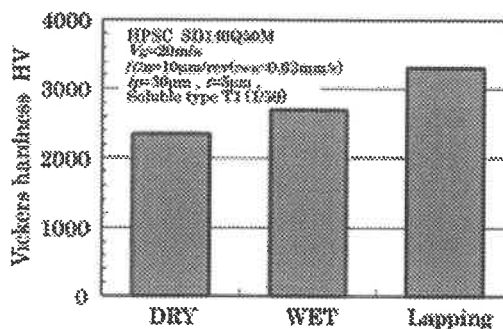


Fig.6 Comparison of Vickers hardness by dry grinding with wet grinding

4.2 Effect of grinding fluid type

In Figure 7, the HPSC surfaces formed using three types of grinding fluids are compared. The surfaces are different from type of grinding fluid. In case of fluid type T1, the ultra-smoothness surface without grinding cracks is observed. In cases of fluid type T2 and T3, the grinding crack is observed on the ground workpiece surface. From the results, it is referred that the suitable grinding fluid is important to be selected for the ultra-smoothness grinding of fine ceramics.

4.3 Effect of grinding fluid dilution

The surface roughness of HPSC ceramic ground by ultra-smoothness grinding using the grinding fluid diluted at 10 times and 50 times is compared in Figure 8. The measuring area is 256 $\times$ 256 μ m². The grinding condition

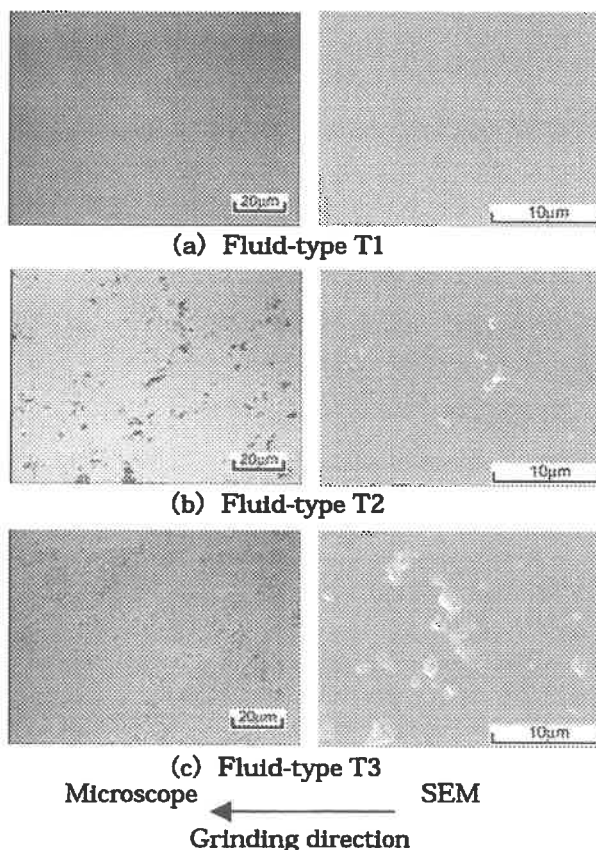


Fig.7 Microscopic photographs of HPSC surface ground at dry grinding and three kind of grinding fluids

Ultra-smoothness grinding $V_g=20$ m/s, $f_{Gn}=10\mu$ m/rev ($f_{wn}=0.63$ mm/s) $t_p=30\mu$ m $t=5\mu$ m, Soluble type $\{1/50\}$

is shown in the Figure. From the result, the 3D surface roughness is almost the same smoothness of about 30nm(Rz) or about 2.4nm(Ra) in both of 10 times and 50 times dilution.

Figure 9 shows the relationship between surface roughness and dilution degree of grinding fluid. From the figure, the surface roughness for the grinding fluid of the dilution of 10 to 50 times is almost the same value. In case of the silicon carbide ceramic, the dilution degree of grinding fluid of 10 to 50 times is considered to have little influence on the surface roughness in ultra-smoothness grinding.

5. Summary

The effect of the grinding fluid supply on ultra-smoothness grinding of silicon carbide ceramic is examined. The main results obtained are as follows:

- (1) Grinding fluid supply is necessary for the ultra-smoothness grinding of silicon carbide ceramic. In case of dry grinding, the surface topography like melted surface layer due to thermal softening is observed on the workpiece surface.
- (2) Grinding fluid type has the strong influence on the ultra-smoothness grinding of silicon carbide ceramic. It is important to select the suitable dilution of grinding fluid for obtaining the ultra-smoothness surface.
- (3) The dilution of grinding fluid has the influence on the surface roughness in ultra-smoothness grinding of silicon carbide ceramic.
- (4) The 3D and 2D surface roughnesses of silicon carbide ceramic formed using the newly developed ultra-smoothness grinding method and the suitable grinding condition attain about 30nm(Rz) or 2.5nm(Ra), and about 10nm(Rz) or 1.5nm(Ra), respectively.

Acknowledgement

The authors would like to thank Mr. Y.Oyama, Y.Tanaka and K.Sonomura. for the assistance to experiments and Noritake Super Abrasive Industries Co. Ltd., Yushiro Chemical Industry Co. Ltd., and Nippon Tungsten Co. Ltd. for providing grinding wheels, grinding fluids and workpiece materials.

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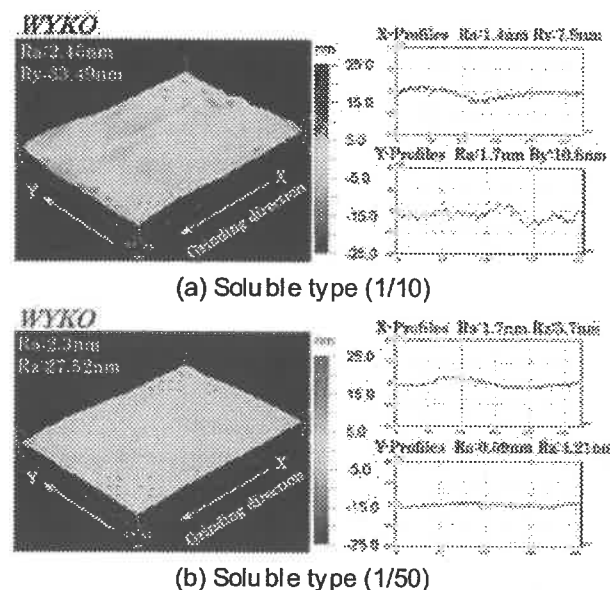


Fig.8 WYKO 3D image and 2D profiles of the HPSC surface ground by ultra-smoothness grinding
Ultra-smoothness grinding [$V_g=20\text{m/s}$, $f_{cn}=10\mu\text{m}$
($w_n=0.63\text{mm/s}$), $f_p=10\mu\text{m}$, $t=5\mu\text{m}$]

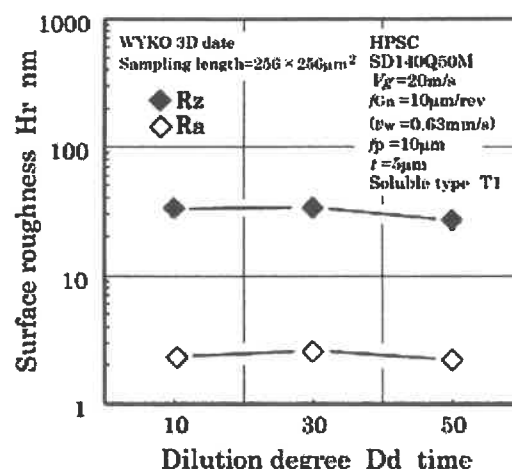


Fig.9 Relationship between surface roughness and dilution of grinding fluid

Quantitative Analysis of the Cutting Edge Behavior on Wheel Surface in Long Grinding Process

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1. Introduction

The grinding wheel surface condition characterized mainly by the distribution and shape of cutting edges has strong influence on grinding results such as surface roughness, grinding force and temperature, and so on. There have been lots of attempts until now, in which the quantitative evaluation of the wheel surface condition has been done. However it is not easy because of the difficulty of quantitative measurement of numerous cutting edges more than several thousands, which distributes irregularly on the wheel surface. In our previous research¹⁾, the automatic image processing system of cutting edges on the wheel surface which is attached to a surface grinder is newly developed for the evaluation of the distribution and wear of the cutting edges. In the system, the evaluation is done precisely by means of measuring and digitizing the numerous microscope images of cutting edges on a circumference of the narrow wheel width to axial direction of wheel spindle. But the system can't evaluate automatically at the same time over the whole wheel surface because of no facility of moving mechanism of the microscope to axial direction of wheel spindle.

In this research, first of all, the moving mechanism of the microscope to axial direction of wheel spindle is developed and attached on the surface grinder. The software program of image processing, furthermore, is built additionally. Using the improved system, the behavior of the same cutting edges on CBN wheel surface in long grinding process from initial grinding process to wheel life is pursued and examined quantitatively. In addition, the relationship between the grinding characteristics and the wheel surface condition is also investigated.

2. Automatic image processing system

Figure 1 shows the schematic view of the automatic image processing system. The system consists of four components, that is, (1) the surface grinder having the wheel spindle and the moving mechanism of the microscope to axial direction of wheel spindle, which are driven with the servo-motor positioned in the accuracy of below $20\mu\text{m}$ and $1\mu\text{m}$ under the computer control, respectively, (2) the observation system of wheel surface using the image transformation technique of CCD camera to the computer for digital handling of microscope image, (3) the software program for extracting of cutting edges from the image transformed into the computer and (4) the whole system control computer operated by multi-task Windows-OS.

The digital image processing is done in the three steps. In the first step, the brilliant image is built by extracting the lighter pixels than the brilliance of threshold from the digital image obtained using the video-capture-board. The brilliant image includes the part of the bright bond surface together with the wear-flat area on active grains. In the second step, the outline of the digital image

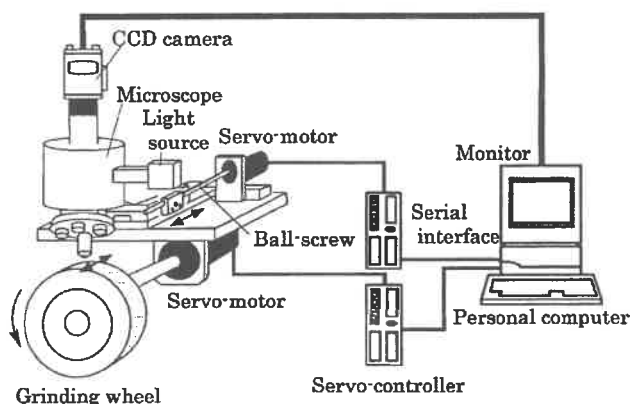


Fig. 1 Schematic view of the automatic image processing system

from the video-capture-board is lined by calculating the difference of the neighborhood pixels. In the final step, the image of the cutting edges are built by overlapping the brilliant image and the outline.

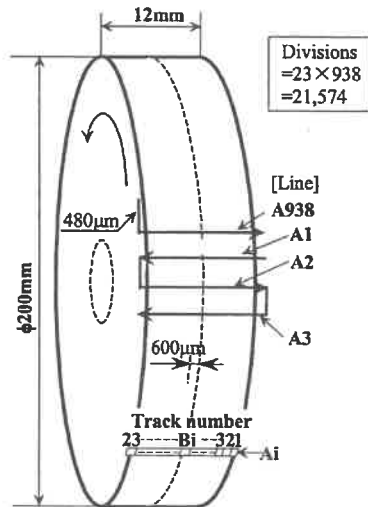


Fig. 2 Division of the wheel surface for the evaluation of surface condition

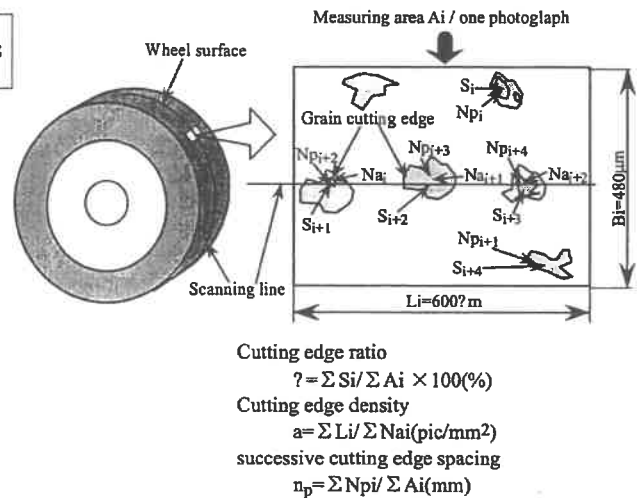


Fig. 3 Parameters for the evaluation of wheel surface condition

3. Experiments

3.1 Evaluation of wheel surface conditions

In order to evaluate the wheel surface conditions precisely, the wheel surface is divided, as shown in Fig. 2, into 23 tracks to axial direction of wheel spindle and 938-location along the circumferential direction of wheel. One photograph image has the dimensions of 600μm in axial direction and 480μm in circumferential direction. The whole wheel surface of φ200mmx12mm used in the experiment consists of 21,574 images. The evaluation of wheel surface condition is done using the parameters such as cutting edge density n_p , cutting edge ratio η and successive cutting edge spacings α , as shown in Fig. 3

3.2 Experimental procedure

The experiments are carried out by plunge grinding method with a surface grinder. The experimental conditions are summarized in Table 1. One grinding pass corresponds to grinding once the workpiece of 100mm long. The observation and roughness measurement of the ground workpiece surface are done with Nomarski interference microscope and a ordinal profilometer. Attaching the apparatus to the profilometer, which can correspond the measuring direction of surface roughness to the

Table 1 Experimental conditions

Wheel	CBN140Q100M $D_o \times B \times D_i: \phi 200 \times 12 \times \phi 50\text{mm}$
Workpiece	SKD11(Hv $\approx$ 800) $L_w \times B_w: 100\text{mm} \times 10\text{mm}$
Wheel speed	$V_g=30\text{m/s}$
Table speed	$v_w=1.8\text{m/min}$
Depth of cut	$t_f=5\text{mm/pass}$
Coolant	Soluble(1/50), Flow rate: 12l/min

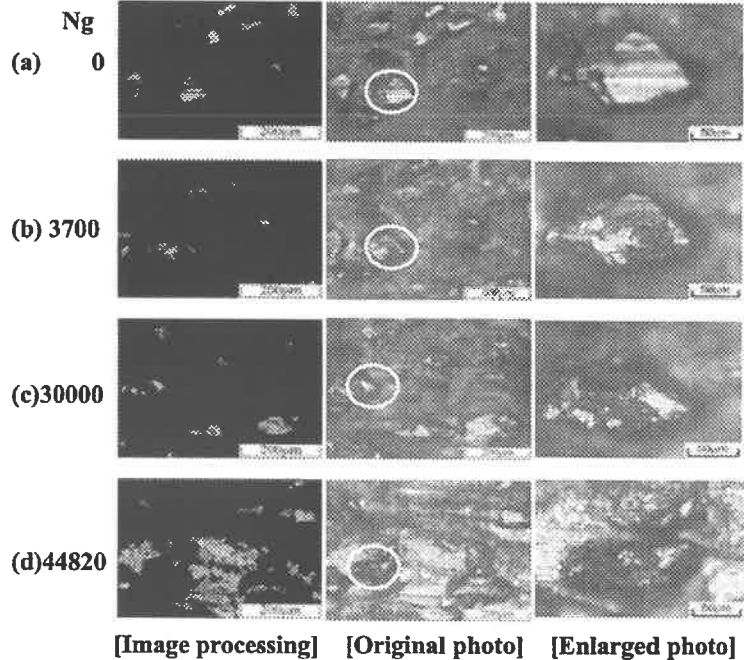


Fig. 4 Variation of wheel surface condition with grinding process [CBN140Q100M, SKD11(Hv=800), $V_g=30\text{m/s}$, $v_w=1.8\text{m/min}$, $t_f=5\text{mm/pass}$, Soluble(1/5)]

grinding direction, the surface roughness is measured for the directions normal and parallel to grinding direction.

4. Results and discussions

Figure 4 shows a typical example of the variation of wheel surface condition with grinding process, including original photographs and cutting edge images extracted by automatic image processing. In the figure, after dressing, in other word, before grinding, the cutting edge has large wear area because the dressing is done by grinding the silicon carbide stick. From the results, it is found that the cutting edge density and the wear area of cutting edge decrease considerably in the initial grinding process. After the initial process, the wheel condition keeps steadily almost the same situation. When grinding until long process of $N_g=44820$, however, the cutting edge density and the wear area of cutting edge become remarkably large. This is why bond wear and loading occur and shine strongly brilliant.

Using the digital cutting images extracted by image processing, the behavior of cutting edges over the circumferential length at a given track in grinding process is shown in Fig. 5. From the results, it is easy to grasp the variation of cutting edge density as mentioned above.

Figure 6 shows the quantitative variation of cutting edge density, cutting edge ratio and successive cutting edge spacing in grinding process. The data are average values for the cutting edges over the whole wheel surface. Generally the cutting edge density and cutting edge ratio are within the range of 30 to 10 cutting edges per mm^2 and of 3 to 0.5 percent, respectively. From the results, the number of cutting edges on the whole grinding surface is over seventy five thousands. The average wear area of cutting edge is about $500 \mu\text{m}^2$, which equals approximately to the area of $23 \mu\text{m} \times 23 \mu\text{m}$ in square shape.

In the initial grinding process until the grinding passes of about 3000, the cutting edge density and the cutting edge ratio decrease rapidly. On the other hand, the successive cutting edge spacing increases steeply. From the results, it is referred that a lot of cutting edges are fractured largely and/or removed from the bond in the initial process. After the initial process, the wheel surface condition keeps steadily at almost the same situation. However, over the long process of $N_g=30000$ which is denoted as the critical grinding process, the cutting edge density and the cutting edge ratio increase rapidly due to the steep increase of bond wear and loading as mentioned above.

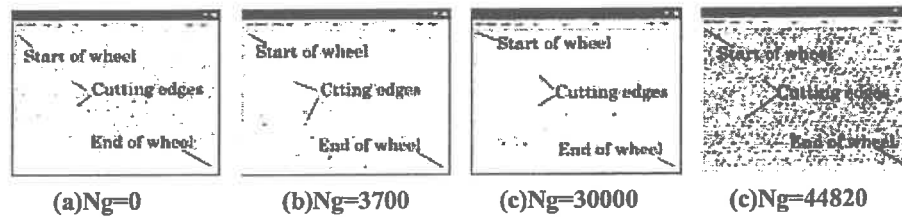


Fig.5 Behavior of cutting edges on a given track in grinding process
[CBN140Q100M,SKD11(Hv=800), $V_g=30\text{m/s}$, $v_w=1.8\text{m/min}$, $t_s=5\text{mm/pass}$,Soluble(1/5)]

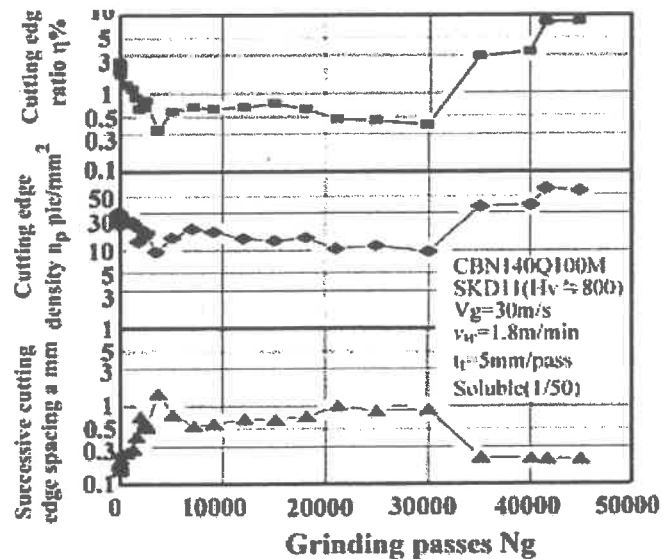


Fig.6 Variation of cutting edge density, successive cutting edge spacing and cutting edge ratio in grinding process

Figure 7 shows the variation of surface roughness with grinding process. From the result, it is found that the surface roughness normal to grinding direction keeps at almost the same value of $5\mu\text{m(Rz)}$ all over the grinding process. On the other hand, the surface roughness parallel to grinding direction decreases steeply from about $2\mu\text{m(Rz)}$ just after initial grinding to $0.5\mu\text{m(Rz)}$ in about $N_g=3000$. The steep decrease is considered to be dependent on the large decrease of loading which occurs due to the dense cutting edge density and large cutting edge ratio just after dressing. However it is noticed

specially that the surface roughness is influenced little by the large variation of wheel surface condition after grinding pass of about $N_g=30000$.

Figure 8 shows the change of the workpiece surface due to the large variation of wheel surface conditions after the grinding pass of about $N_g=30000$. In the grinding pass of $N_g=30000$, the workpiece surface is the normal grinding surface. In the grinding pass of $N_g=44820$, however, the occurrence of grinding crack is observed on the workpiece surface. The occurrence is considered to depend on the large grinding force and grinding heat caused by the large cutting edge ratio.

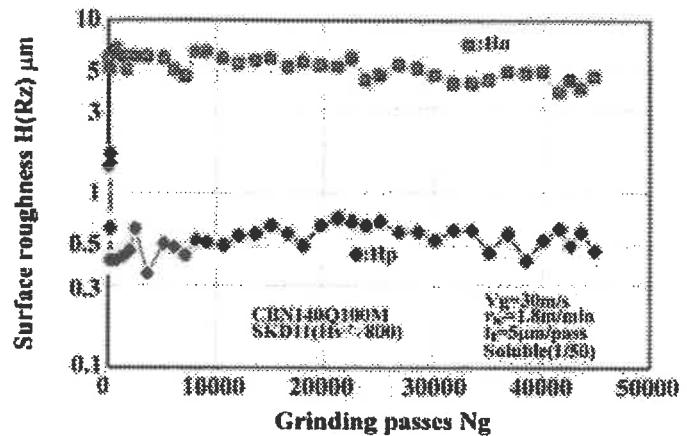
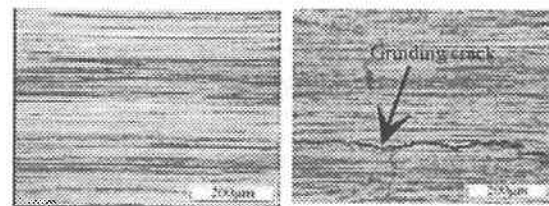


Fig.7 Variation of the surface roughness in grinding process



(a) $N_g=30000$

(b) $N_g=44820$

Fig.8 Occurrence of grinding crack due to the increase of cutting edge ratio by grinding
[CBN140Q100M,SKD11(Hv=800), $V_g=30\text{m/s}$,
 $v_w=1.8\text{m/min}$, $t_g=5\text{mm/pass}$,Soluble(1/5)]

4. Summary

The main results obtained are as follows:

- (1) The improved automatic image processing system is useful for evaluating automatically the behavior of numerous cutting edges more than seventy five thousands over the whole wheel surface in long grinding process.
- (2) In the initial grinding process, a part of cutting edges on the CBN wheel surface is fractured largely and/or removed from the bond. After the initial process, the wheel condition is almost the same until the critical grinding process. Over a critical grinding process, the cutting edge density and cutting edge ratio increase steeply due to the remarkable occurrence of the bond wear and loading.
- (3) The behavior of cutting edges in grinding process has little on the surface roughness in the grinding process instead of the initial grinding process. From the excessive increase of the cutting edge ratio due to the long grinding process, the grinding crack occurs on the workpiece surface.

Acknowledgement

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Grinding Wheel Loading with and without Vibration Assistance

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Introduction

Due to its tool structure and composition, the grinding operation is considered the best machining process in the manufacture of brittle materials like ceramics. A grinding wheel distributes wear over many cutters and is capable of replenishing itself. However, the condition of the grinding wheel constantly changes as grits wear and chips accumulate in the spaces between grits. Therefore, wheel loading is one of the main factors limiting the capability of the grinding wheel. Chip accumulation (or loading) is particularly problematic with the fine grit wheels used in fine grinding of ceramics. Wheel loading results in an increase in grinding forces and temperature. As a consequence, the rate of abrasive wear increases and the surface finish of the workpiece deteriorates.

In our previous studies,^[1] we have investigated the effect of wheel characteristics, such as concentration and grit size, on the grinding chip size. The literature indicates that the main cause of chip (or wheel) loading is adhesion between the active grits and the workpiece material deformed in the form of chips.^{[2][3]} Chips accumulate in the spaces between grits. Thus the relationship between chip volume and volume between grits is likely to influence the tendency for wheel loading. These quantities were investigated using a kinematic based simulation. Based on the grinding conditions and wheel characteristics, the simulation program calculates chip volumes. A higher ratio of $V_{\text{between grits}}/V_{\text{chip}}$ would reduce the tendency for chip accumulation. Our results indicate that the volume between grits grows more quickly than the volume of chips as the grit size increases and/or volume fraction decreases. This validates the common observation that fine grit wheels experience more chip loading than coarse grit wheels. In order to gain a better understanding of the mechanism of chip loading, we continued the study by examining the effect of vibration assistance.

Vibration Assisted Grinding

Vibration assisted grinding is a minor modification to the conventional grinding process, which results in enhanced performance and economic efficiency. Research on vibration assisted grinding has shown that the superposition of high frequency vibration to the regular grinding process produces changes in the forces, tool wear, surface quality, and subsurface properties.

During our study of grinding wheel loading, a large difference was observed in the wheel loading results between the conventional grinding process and the vibration assisted grinding process. Therefore, these two grinding processes were investigated separately. The goals were to find the advantages of vibration assisted grinding over grinding without vibration and the effects of vibration frequency and amplitude on grinding wheel loading.

Method of Qualifying Wheel Loading Condition

There are many methods to measure wheel loading condition, including chemical detection, calorimetry, spectroscopy, eddy current sensing, magnetization, radiotracing, and x-ray fluorescence.^[4] We have adopted a simple approach for quantifying wheel loading that utilizes microscope images and image analysis software. A microscope was used to capture wheel surface images along the wheel circumference. Then Scion Image software was used to determine the percentage of chip loading. Figure 1 shows a typical image as acquired and after conversion to black-and-white. The black area represents the chip area, and its percentage can be calculated by the software. In our test, we took seven images around the wheel, analyzed each image, and used the average percentage value as the chip loading condition for each situation.

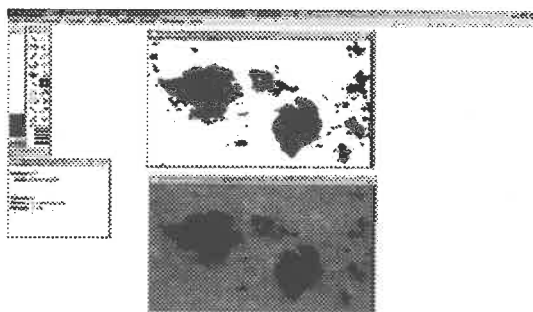


Figure 1 Raw Image (bottom); Black and White (top)

Effect of Vibration Assisted Grinding

Two types of aluminum oxide grinding wheels were used to study the effects of vibration assisted grinding on wheel loading (Norton 38A80-JVBE and 38A80-IVBE). A narrow steel workpiece was surface ground. After each table cycle, the depth of cut was indexed. Figure 2 and 3 show the experimental results. In both cases vibration assisted grinding reduces of wheel loading. In the following section, we will look more closely into the effects of two vibration parameters: frequency and amplitude.

Vibration Frequency Effect

The first test was done with the Norton 38A80-HVBE grinding wheel and a 12.7mm x12.7mm x 50.8mm 1018 steel workpiece. The grinding depth of cut was 0.025mm and the table feedrate of the grinding machine was 65mm/s. The vibration amplitude was 5 μ m, and the three vibration frequencies were 0.5kHz, 1kHz and 2kHz. The results are shown in Figure 4. Before about seventy passes, there is no obvious wheel loading difference between grinding with and without vibration. But after seventy passes, the effect of vibration becomes more obvious. Wheel loading tends decrease with increased vibration frequency.

In order to confirm this, another test was done with the same type of workpiece but with the Norton 38A80-IVBE grinding wheel and different grinding conditions. The grinding depth of cut was 0.020mm and the table feedrate was 50mm/s. Vibration amplitude of 5 μ m and vibration frequency of 1kHz and 2kHz were used. (See Figure 5). These results also show that higher vibration frequency should be recommended in order to get lower wheel loading condition.

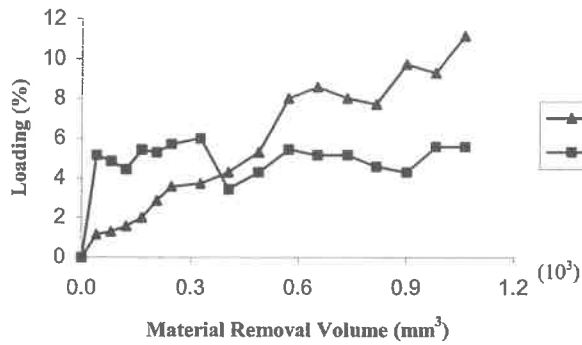


Figure 2 Vibration Effect-Test 1

Wheel model: 38A80-JVBE
 Workpiece: 12.7mm x 12.7mm x 50.8mm 1018 Steel
 Table speed = ± 9 mm/s
 Wheel rpm=2907rpm
 Depth of cut=0.025mm
 No coolant
 $f=2$ kHz
 $A=5\mu\text{m}$

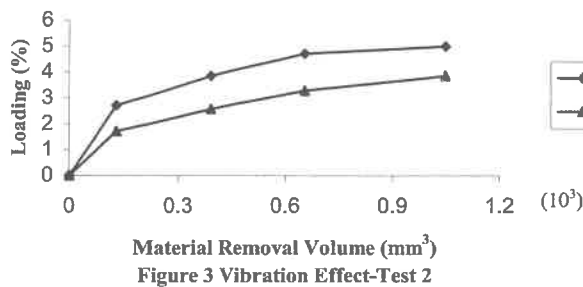


Figure 3 Vibration Effect-Test 2

Wheel model: 38A80-IVBE
 Workpiece: 12.7mm x 12.7mm x 50.8mm 1018 Steel
 Table speed = ± 50 mm/s
 Wheel rpm=2907rpm
 Depth of cut=0.020mm
 No coolant
 $f=2$ kHz
 $A=5\mu\text{m}$

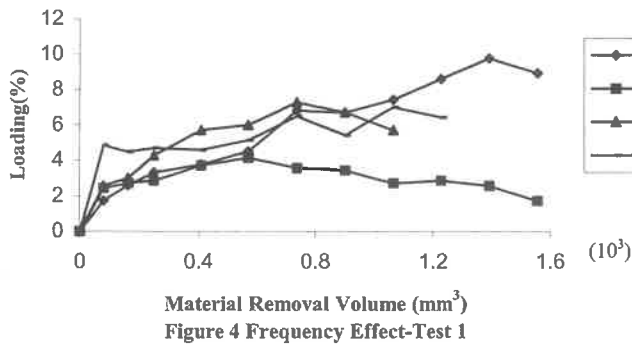


Figure 4 Frequency Effect-Test 1

Wheel model: 38A80-HVBE
 Workpiece: 12.7mm x 12.7mm x 50.8mm 1018 Steel
 Table speed = ± 65 mm/s
 Wheel rpm=2907rpm
 Depth of cut=0.025mm in
 No coolant

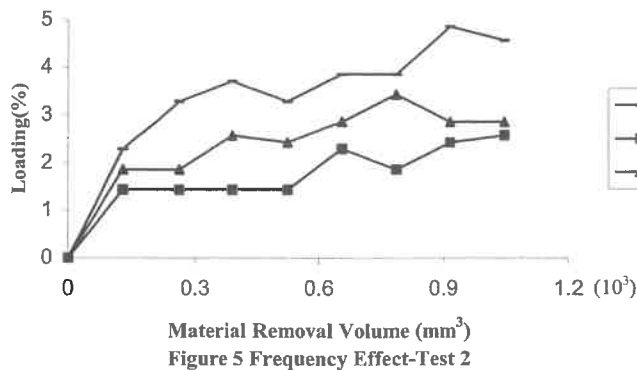
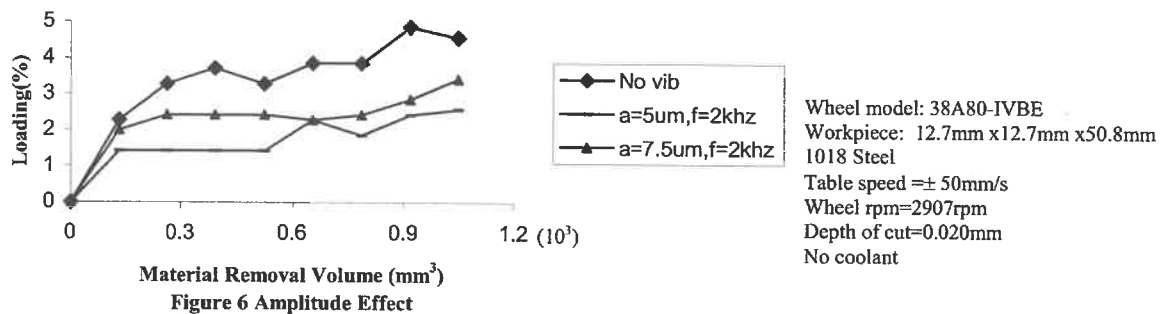


Figure 5 Frequency Effect-Test 2

Wheel model: 38A80-IVBE
 Workpiece: 12.7mm x 12.7mm x 50.8mm 1018 Steel
 Table speed = ± 50 mm/s
 Wheel rpm=2907rpm
 Depth of cut=0.020mm
 No coolant

Vibration Amplitude Effect

Figure 6 shows the results of tests at different vibration amplitudes. The workpiece was again 1018 steel and the wheel model was Norton 38A80-IVBE. The grinding depth of cut was 0.020mm and the table feedrate was 50mm/s. The experimental results show that amplitudes of 5 μ m and 7.5 μ m both result in less loading than an amplitude of 0 (no vibration case). However, the 7.5 μ m amplitude produces no benefit beyond the 5 μ m tests. This may indicate there is a threshold beyond which no additional benefit occurs.



Conclusions

Besides its effects on forces, tool wear, surface quality, and subsurface properties, vibration assisted grinding was found here to be a way of reducing wheel loading. Experimental results show that as vibration frequency increases wheel loading decreases. There is likely a threshold beyond which higher frequencies and amplitudes would produce no additional advantage. For future research, we will identify this threshold. In addition, we will investigate how wheel loading affects grinding force. We will also look into the relationship of wheel loading and material properties by checking the chip shape with SEM.

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CLOSED LOOP CONTROL OF TOOL DEFLECTION

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INTRODUCTION

Milling with small diameter cutters can produce errors in the shape of features machined on hard materials. Typically a small diameter is needed to reach down into complex features and the large difference between the radial and axial stiffness of the tool will create form errors in the part. The goal of this research is an automatic way to compensate for these errors and improve the quality of the machined components.

While an open loop deflection compensation algorithm [1] requires a cutting force model, the closed loop technique involves deflection compensation based on measured force [2,3]. There are two major quantities that must be known, force and stiffness. The idea is to measure the cutting force in real-time, predict a deflection based on the stiffness measurements and compensate with a piezoelectric (PZT) actuator that can move the spindle [3]. Two methods of measuring force were explored. The first method involved measuring the Z-component of the machining forces on the workpiece and the second method involved measuring the axial component of the tool force on the spindle.

Figure 1 shows the orientation of the spindle with respect to the workpiece. Notice the axial spindle force (Y'-direction) is equivalent to the Z-workpiece force times the cosine of the tilt angle, α . The spindle is mounted on the slide of the DTM and can move in the Z-direction. Tool deflection occurs in both the Y' and Z' directions, while the PZT actuator can only move in the Z' direction. However, both directions of deflection can be compensated, because the tool tilt angle is known.

FORCE MEASUREMENT ON WORKPIECE

Control Algorithm Equation (1) is the control algorithm used with this technique. It shows how the force measurement is turned into a position error using the tool stiffness. The "desired position" for the spindle mounted error compensation technique is zero. So for the error in Equation (1) to be zero, the tool deflection (force/stiffness) must be compensated by PZT motion. The result is a tool that appears rigid and errors from tool deflection are eliminated from the part.

$$error = position_{desired} - position_{actual} + \frac{force}{stiffness} \quad (1)$$

There are two control loops: the inner loop provides feedback from the capacitance gage used as position feedback for the PZT actuator and the outer loop takes the measured force and transforms it to estimate total tool deflection.

Both radial and axial components of tool deflection can be compensated by motion of the PZT actuator on the spindle. Figure 2 shows the tool and the components of the workpiece force (F_z) broken into components normal to ($F_{z'}$) and along ($F_{y'}$) the spindle. It also shows the deflection in the Z' and Y' directions and δ_{PZT} - the scaled PZT displacement needed to correct the tool deflection normal to the

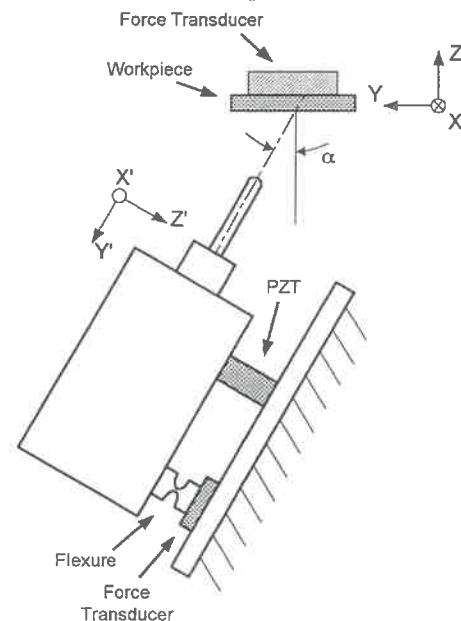


Figure 1. PZT-Actuated Spindle [3]

workpiece. An expression for the PZT deflection needed to correct the tool deflection as a function of the force, stiffness and deflection is given in Equation (2).

$$\delta_{PZT} = \frac{F_Z \left[\frac{\sin^2 \alpha}{k_R} + \frac{\cos^2 \alpha}{k_A} \right]}{\sin \alpha} \quad (2)$$

EXPERIMENTAL RESULTS

Experimental tests were conducted on S7 steel with a spindle speed of 10,000 rpm. The desired groove profile consists of a linearly varying depth from 0-80 mm over a length of 20 mm. A plot of raw z-force from the workpiece force transducer that was captured during a 20° tool tilt experiment is at the left in Figure 3. Note that the force does not smoothly increase as expected with the linear increase in depth of cut. A peak and hold algorithm was implemented to extract the peak force for each tool rotation. Based on this force, an effective radial deflection (right side of Figure 3) was calculated at each time step based on tool tilt, stiffness, and current tool force and was used in the control algorithm, Equation (1), to generate a new voltage command to the PZT. Notice the maximum radial deflection (δ_{PZT}) is approximately 40 μm , which corresponds to a Z-direction error (δ_z) on the part of 14 μm (see Figure 2).

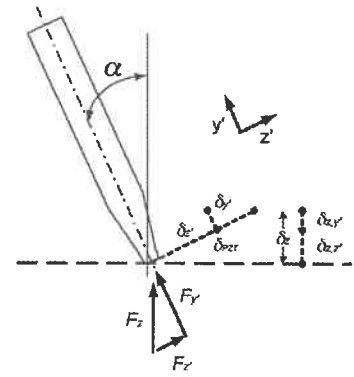


Figure 2. Forces on tilted tool

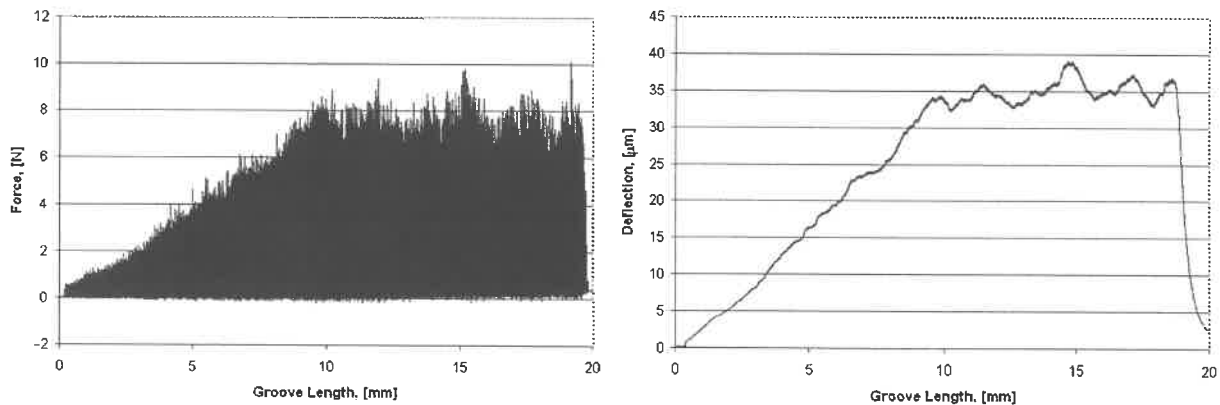
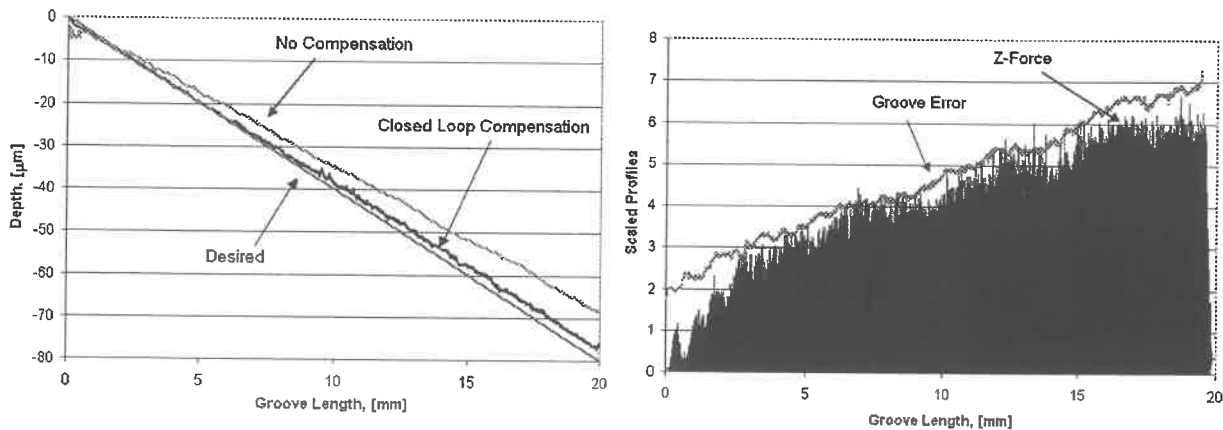


Figure 3. Raw Z Force (left) and Effective Radial Deflection from 20° Tilt Closed Loop Experiment; Depth = 0-80 mm, Upfeed = 10 mm/rev, Spindle Speed = 10,000 rpm, 4 mm long tool shank

Groove profiles measured by the Talysurf profilometer (Figure 4(a)) illustrate the improvement in the shape of the groove for the closed loop compensation when compared no compensation. Without compensation the error is on the order of 12 μm and this becomes less than 4 μm with compensation. The limit for creating the desired shape can have a variety of sources. The feedback control system did an excellent job keeping the error in Equation (1) less than 500 nm. As a result, the accuracy of the groove shape will be a function of uncertainty in 1) measurement of the groove, 2) force measurement, 3) transducer calibration, 4) tool stiffness measurement and 4) the touch off procedure used to find the part and begin the experiment.

The groove shape is measured by dragging a Talysurf probe down the center of the groove with the overall depth dependent on leveling the trace with the start and end points. The test pieces are ground flat so the surface of the part is assumed to be straight between the start and end points. To evaluate the validity of this assumption, the measured normal force and shape of an uncompensated groove with the tool tilted at 20° is shown in Figure 4(b). Notice the depth of the groove generally (but not exactly)

followed the force over the 5-minute duration of the experiment, but that neither was linear. Since there was no force feedback and a linear increase in depth was commanded, the nonlinear force and groove shape has not been explained. It could be a result of a non-flat part, material variations, or other effects. These unknowns may effect may limit the capability of this technique to produce a perfect part.



a) Shape of Groove for compensated and uncompensated experiments

b) Uncompensation Groove Profiles and normal force vector

Figure 4. Force and groove profile for tool tilted at 20°

Depth = 0-80 mm, Upfeed = 10 mm/rev, Spindle Speed = 10,000 rpm, 4 mm long tool shank

FORCE MEASUREMENT ON SPINDLE

Measuring forces on the spindle offered several advantages over measuring forces on the workpiece. One advantage is the opportunity for a self-contained apparatus that could measure forces and correct for tool deflection. This allows tool deflection algorithms to be implemented independent of workpiece properties such as size and mass. Another advantage of a self-contained design is that the force can always be measured along the radial and axial direction of the tool, negating the need to know the tool tilt angle.

Spindle Force Calibration A three-axis force transducer of the same type used to support the workpiece was built into the spindle. However, the location of this transducer made prediction of the tool forces more complicated and a calibration procedure was employed. The relationship between forces acting on the tool tip and forces measured by the transducer was measured by applying known loads to the tool tip and recording the measured forces in each direction on the transducer. Figure 5 is a schematic illustrating the calibration technique both in the axial (A) and radial (B) directions.

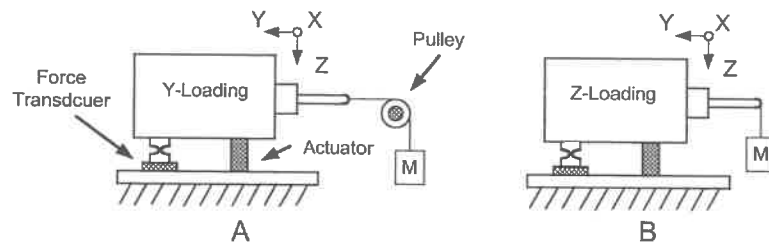


Figure 5. Schematic of Static Calibration Technique

A linear trend line was used to relate the tool-tip force to the measured transducer force in each direction. The coefficient matrix of Equation (3) was then assembled from the force ratio plots. It should be noted that the force transducer measures a coupled effect from Y and Z-tool forces. This is due to the moment that results from an axial load. The matrix in Equation (3) relates the L_k (measured forces in the x, y, and z directions) to the F_k (forces recorded in each direction on the load cell).

$$\begin{bmatrix} L_x \\ L_y \\ L_z \end{bmatrix} = \begin{bmatrix} -0.50 & 0 & 0 \\ 0 & 0.44 & 0 \\ 0 & 0.80 & -1.91 \end{bmatrix} \begin{bmatrix} F_x \\ F_y \\ F_z \end{bmatrix} \quad (3)$$

This matrix can be inverted to find the forces applied to the tool based on the load cell readings and used for compensation. One complication of this design is crosstalk between the PZT actuation and measurement systems. When the actuators are extended, the force transducer records a reduction in force. This is not ideal, but repeatable measurements made it possible to produce a second calibration curve, which was added to the control algorithm to compensate for this effect.

Control Algorithm The control algorithm used with the spindle-mounted force transducer was somewhat different than the algorithm used with the workpiece transducer because of complications of the spindle force measurement. The major difference is the capacitance gage signal fed into the force signal to compensate for the actuation/force effects discussed above.

Experimental Results Experimental cuts were made in S7 steel with 30° tool tilt using the feedback algorithm. Figure 6 shows the compensated groove with spindle force feedback as well as a groove without compensation. The groove profile compared well with the desired 0-80 mm depth over the groove span of 20 mm. Notice that the spindle feedback groove overcompensated at the start of the cut (4 μm error) but ended at the desired depth of 80 μm. Over most of the groove, the error was less than 2 μm.

X.3 CONCLUSIONS

Force feedback using either the workpiece or the spindle mounted force gage improved the fidelity of the grooves machined in S7 steel samples. The error was reduced from 16 μm to less than 2 μm over the 80 μm deep, 20 mm long grooves. The limit of the correction using this technique has not been determined but nonlinear effects between force and deflection will ultimately set the best possible compensation.

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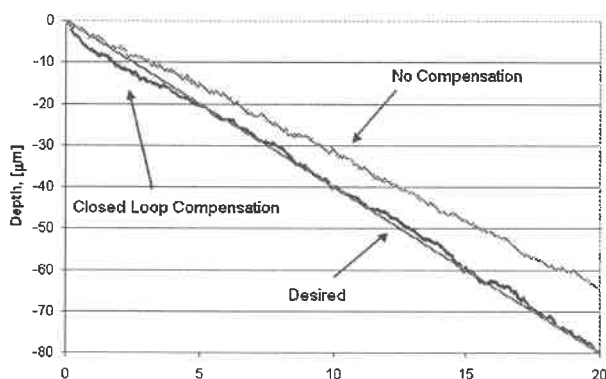


Figure 6. Desired, Spindle compensation and No Compensation Profiles, Tilt = 30°, Depth = 0-80 mm, Upfeed = 10 mm/rev, Spindle Speed = 10,000 rpm, 4 mm long tool shank

An Investigation of the Dynamic Absorber Effect in High-Speed Machining

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Introduction

Recent improvements in machine and spindle designs have led to the increased use of high-speed machining (HSM) in the manufacture of discrete parts, especially in the aerospace industry [1]. HSM seeks to increase depth of cut, and the corresponding material removal rate (MRR), through the selection of optimum cutting parameters. It is recognized that a major practical limitation on the productivity of HSM systems is regenerative chatter [2]. One method of pre-process chatter prediction and avoidance is the well-known stability lobe diagram. Stability lobe diagrams, which predict system stability as a function of selected machining parameters, are can used to select the best available spindle speed for maximized MRR. Stable and unstable regions (separated by the stability “lobes”) are seen in these diagrams, depending on the selected spindle speed and axial depth of cut, b , for peripheral end milling.

The equation for the limiting depth of cut, b_{lim} , at each spindle speed is shown in Eq. 1, where K_s is the specific cutting energy, m^* is the average number of teeth in the cut (which depends on the number of teeth on the cutter and the selected radial immersion), and $\text{Re}[G_{22}(w)]_{\text{Oriented}}$ is the negative portion(s) of the real part of the oriented tool point frequency response function (FRF) [2]. As seen in Eq. 1, b_{lim} can be increased by minimizing the value of $\text{Re}[G_{22}(w)]_{\text{Oriented}}$, resulting in increased depths of cut and higher MRR.

$$b_{lim} = \frac{-1}{2K_s \text{Re}[G_{22}(w)]_{\text{Oriented}} m^*} \quad (1)$$

This work describes a method to shift the stability lobes up to increased depth of cut levels by decreasing the minimum value of the negative real part of the tool point FRF. This is achieved by matching the fundamental frequency of the cantilevered tool to a natural frequency of the holder/spindle, similar to the changed assembly response observed when implementing a damped dynamic absorber. In this work, the ‘dynamic absorber effect’ is verified experimentally with simple flexures. A model of a lumped-mass, two degree-of-freedom (DOF) system, created through receptance coupling techniques applied to two single DOF systems, is used to predict the response at the top of the flexure assembly as the ratio between the top flexure and bottom flexure mass and stiffness is modified.

Flexure Designs

The 2DOF flexure shown in Figure 1, produced by assembling two single DOF flexures, was used to experimentally validate the ‘dynamic absorber effect’. Three flexures were produced for testing purposes, a single top flexure, a large base flexure, and a small base flexure. Based on available tooling and material, the three flexures were designed for particular natural frequency values [3]; results are shown in Table 1. For the case when it was required to match the natural frequencies of the single DOF modes, additional mass was added to the base flexures to lower the natural frequency to approximately 300 Hz, the cantilevered natural frequency of the top flexure.

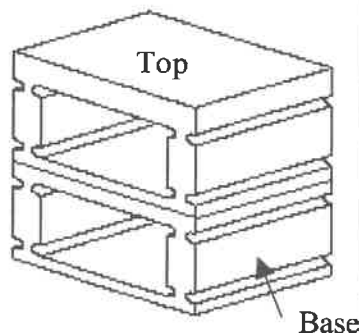


Figure 1: 2DOF flexure.

Table 1: Flexure design parameters and results

	m (kg)	k (N/m)	c (kg/s)	W_n (Hz)
Large Base Flexure				
Flexure Theory Prediction	1.54	1×10^7	-	405.6
Modal Testing Results	1.43	9.41×10^6	46.9	408.3
Modal Testing Results With Mass	2.46	8.85×10^6	70.6	301.9
Small Base Flexure				
Flexure Theory Prediction	0.146	1×10^6	-	416.5
Modal Testing Results	0.108	6.34×10^5	4.18	385.0
Modal Testing Results With Mass	0.155	5.68×10^5	2.23	304.7
Top Flexure				
Flexure Theory Prediction	0.223	1×10^6	-	337.0
Modal Testing Results	0.1454	5.04×10^5	1.17	296.3

After manufacture, the two single DOF base flexures and one single DOF top flexure were modally tested to extract the natural frequency (W_n), mass (m), stiffness (k), and damping (c) values. All flexures, including the top flexure, were adhered to ground with cyanoacrylate and instrumented with an accelerometer. The modal mass, damping, and

stiffness values were extracted from the measured FRFs by fitting the data using a peak picking method [4]. The modal testing flexure theory results are compared in Table 1 for the three flexures.

Flexure Results

Modal testing was performed on the coupled flexures to verify the 'dynamic absorber effect' that takes place when the cantilevered natural frequencies of the base and top flexures are matched. For all testing, ground was defined as a machining tombstone with a large mass and stiffness and all connections were made with cyanoacrylate. Figure 2 displays the real part of the FRF for the large base flexure (connected to ground), the real FRF of the top flexure (connected to ground), and the real FRF of the combined flexures with the large base flexure connected to ground and the top flexure connected to the top of the base flexure. From Figure 2, it can be seen that the natural frequency of the cantilevered base flexure is 408 hertz and the natural frequency of the top flexure cantilevered to ground is 296 hertz. The modes are separated. When the top flexure is connected to the base flexure, the negative real magnitude of the dominant mode of the combined system is 38.5 % less than the negative real magnitude of the top flexure cantilevered to ground, even in the absence of a direct match in the cantilevered SDOF natural frequencies.

For the next set of modal testing, additional weight was added to the base flexure so that the cantilevered natural frequency of the base flexure was lowered to approach the cantilevered natural frequency of the top flexure. Figure 3 displays the real FRF of the modified large base flexure (connected to ground), the real FRF of the top flexure (connected to ground), and the real FRF of the combined flexures with the modified base flexure connected to ground and the top flexure connected to the top of the modified base flexure. From Figure 3, it can be seen that the dominant natural frequency of the cantilevered base flexure is now 302 Hz and the dominant natural frequency of the top flexure cantilevered to ground is again 296 Hz. The modes of the individual cantilevered flexures are close and now interact. *When the top flexure is connected to the base flexure, the negative real value of the dominant mode of the combined system is 69 % less than the negative real value of the top flexure attached directly to ground.* By matching the natural frequencies of the individual cantilevered systems, the dominant mode of the base flexure interacts with the dominant mode of the top flexure. The two modes of the combined system split around the original cantilevered natural frequency and the minimum value of the negative real response of the dominant combined system mode is significantly smaller than the cantilevered response of the top flexure attached directly to ground. The base flexure has acted as a 'dynamic absorber' to decrease the response of the top flexure.

Figure 4 compares the real FRFs for the combined systems when 1) the top and bottom flexure cantilevered modes are separated; and 2) the top and bottom flexure dominant modes have similar natural frequencies. As shown, when the modes have similar natural frequencies, the combined system response is significantly reduced.

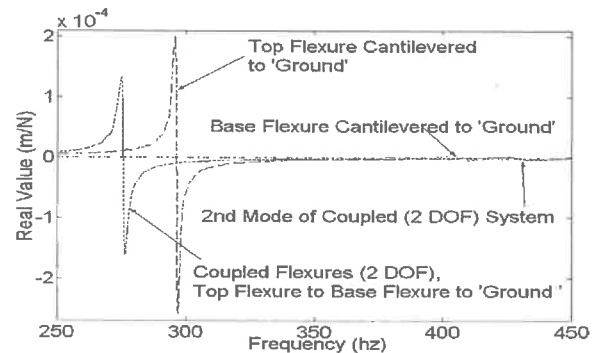


Figure 2: Combined flexure response, initial SDOF modes separated

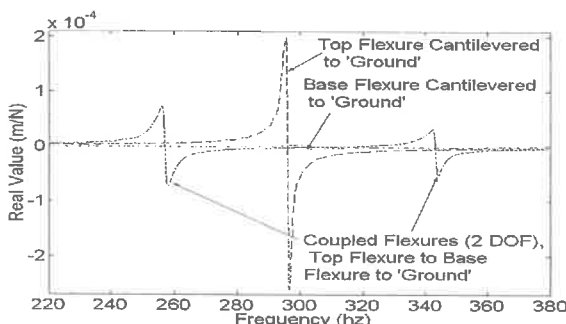


Figure 3: Combined flexure response, initial cantilevered modes together

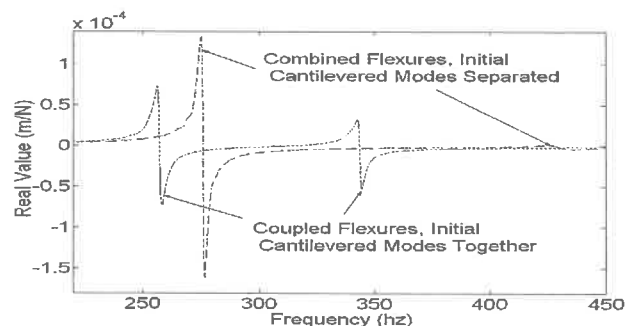


Figure 4: Combined flexure response comparison

Model Development

For the coupled flexure assembly, Figure 5 shows the lumped parameter model that was used to describe the system. The base flexure, system A, was modeled as a single degree-of-freedom (DOF) substructure, defined as a mass, m_1 ,

connected to ground through a spring, k_1 , and a viscous damper, c_1 . System B was modeled as a single DOF substructure with free-free boundary conditions; it consisted of a mass, m_2 , connected to a massless coordinate, x_3 , through a spring, k_2 , and a viscous damper, c_2 . The response of the assembly, system C, at coordinate X_2 (representing the uppermost point on the top flexure) for a harmonic force, F_2 (applied at coordinate X_2) is computed using receptance coupling substructure analysis (RCSA) [5-7] based upon the receptances of the two single DOF substructures. It is assumed that the substructure receptances for the rotational degrees-of-freedom are negligible (by design for flexures) and that the substructures are rigidly connected.

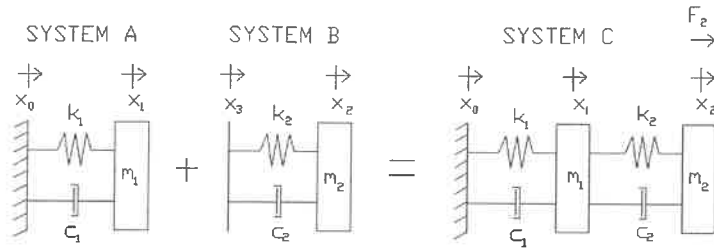


Figure 5: Model of two SDOF systems combined into a 2DOF system

The direct FRF of the assembled system at the top flexure free end, $G_{22}(w)$, shown in Figure 5 was derived using receptance coupling [8]. The resulting equation, expressed in terms of the system A and B mass, damper, and spring constants, is shown in Eq.2, where the frequency, w , has units of radians per second.

$$G_{22}(w) = \frac{X_2}{F_2} = \frac{-1}{w^2 m_2} - \left(\frac{-1}{w^2 m_2} \right) \left[\left(\frac{1}{-m_1 w^2 + i w c_1 + k_1} \right) + \left(\frac{m_2 w^2 - i w c_2 - k_2}{w^2 (i w m_2 c_2 + m_2 k_2)} \right) \right]^{-1} \left(\frac{-1}{w^2 m_2} \right) \quad (2)$$

Predicted Responses

To test the model, the parameters for the mass, dampers, and springs produced from modal testing (Table 1) were substituted in Eq. 2 to solve for $G_{22}(w)$. Figure 6 displays measured and predicted results for $\text{Re}(G_{22}(w))$ for the coupled flexure system with the large base. For this case, the individual flexures had approximately the same natural frequency. Two additional coupled flexure responses were also compared to predicted results to further validate the model. Figure 7 displays results for $\text{Re}(G_{22}(w))$ for the coupled flexure system with the large base and separated individual flexure modes. Figure 8 displays the results for the coupled flexure system with the small base. In this case, the mass and spring values, as determined by modal testing, of the small base were approximately 16 times smaller than the values for the large base, and the individual flexure modes were matched with approximately the same natural frequency. In all instances, the model results approximate the actual results produced from modally testing the coupled, 2DOF flexure system. Discrepancies between actual and predicted results can be attributed to the decision to neglect rotational degrees of freedom and unmodeled flexibility/damping in the connection between flexures.

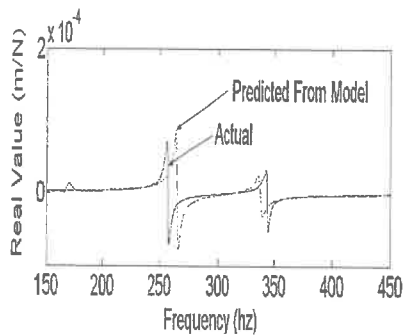


Figure 6: Real $G_{22}(w)$ with matched individual flexure modes and large base flexure

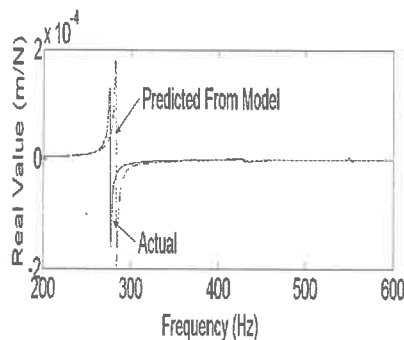


Figure 7: Real $G_{22}(w)$ with separated individual flexure modes and large base flexure

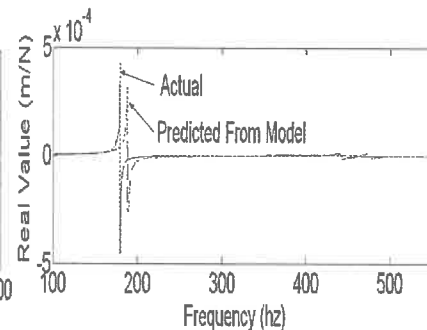


Figure 8: Real $G_{22}(w)$ with matched individual flexure modes and small base flexure

Once the model was validated, the model parameters could then be manipulated to predict $\text{Re}(G_{22}(w))$ coupled flexure results for other cases. Of interest was the $G_{22}(w)$ response when the natural frequencies of the individual base and top flexures remain matched, as shown in Eq. 3, but the ratio between the base and top flexure mass and spring values was increased by a multiplier value MV as shown in Eq. 4, i.e., as MV increases, the base flexure becomes stiffer and more massive relative to the top flexure.

$$W_{n,base} = \sqrt{\frac{k_{base}}{m_{base}}} = W_{n,top} = \sqrt{\frac{k_{top}}{m_{top}}} \quad (3)$$

As shown in Figure 9, which displays the real response of the coupled flexure system, when MV is equal to 1 (the base flexure and top flexure are identical), the coupled flexure system has two modes that split around the individual flexure natural frequency of 300 Hz. As MV increases, the two modes of the coupled flexure system approach each other and, when MV is large enough, the combined response approaches a single DOF. Of particular interest is the fact that the minimum value of the negative real part of the dominant mode decreases with MV until the two modes begin to converge to a single DOF and the value of the negative real part begins to increase again. *In other words, in terms of the value of the negative real part of the dominant coupled system mode, the optimum point is not found by increasing the mass and stiffness of the base flexure ad infinitum.*

Discussion and Conclusions

The results of the flexure testing and modeling can be extended to the case of the coupled spindle/holder and tool in high-speed machining operations. The fundamental tool mode can be considered analogous to the top flexure mode and each holder/spindle mode is analogous to the base flexure mode. As shown in the flexure results, if the fundamental cantilevered tool mode can be matched to a holder/spindle mode, the minimum value of the negative real part of the FRF for the coupled holder/spindle/tool system, as measured at the tool point, can be reduced. The 'dynamic absorber effect' that occurs due to the interaction between the modes reduces the tool point negative real minimum value relative to the response that would occur if the tool was cantilevered directly to ground (i.e., an infinitely stiff holder/spindle). From Eq. 1, by taking advantage of the 'dynamic absorber effect', the value for b_{lim} can therefore be increased. The tool natural frequency can be varied through tool overhang length and/or diameter adjustment, changes in tool material (modulus), or manipulation of the connection between the tool and the holder.

A second conclusion that can be drawn from this testing is that the largest multiplier value, or greatest mass and stiffness of the base flexure, does not lead to a minimized real part of the coupled flexure system at the free end of the top flexure, $\text{Re}(G_{22}(w))$. In terms of a holder/spindle/tool assembly, it can be surmised that during the design of spindles for certain tool geometry ranges, if the interaction between tool and holder/spindle modes is taken into consideration, the stiffest, largest mass spindle may not be the optimum selection for stable machining.

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$$MV = \frac{k_{base}}{k_{top}} = \frac{m_{base}}{m_{top}} \quad (4)$$

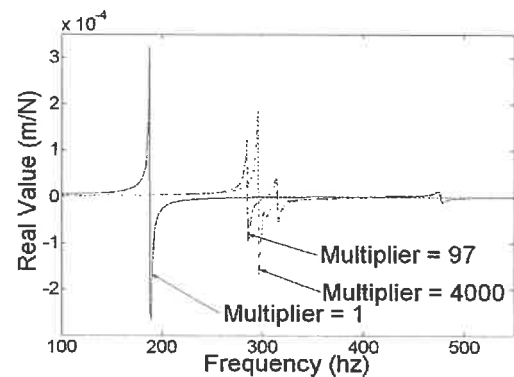


Figure 9: Coupled $\text{Re}(G_{22}(w))$ for $MV = 1, 97$, and 4000

MACHINING OF HARD COATINGS IN OPTICAL QUALITY

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Abstract

Precision molds for hot pressing of optical glasses and polymers require high temperature materials which often do not possess long-term corrosion and sufficient wear resistance. These requirements can be met by dedicated coatings which provide corrosion protection and wear resistance and prevent adhesion between the molded glass and the mold material. However, such coatings, once deposited on the mold surface, must be ground, milled, turned or polished to optical finish [1]. This work deals with the material removal mechanisms of newly developed hard coatings. Two classes of PVD deposited coatings as well as two different types of sol-gel derived coatings have been investigated. While the PVD coatings and sol-gel derived ZrO_2 ormosil coatings can be machined in ductile mode, inorganic sol-gel derived ZrO_2 coatings can only be machined in brittle mode.

Sol-gel derived ZrO_2 coatings

Nano-crystalline ZrO_2 layers can conveniently be deposited by the so-called sol-gel process. X40Cr13 steel substrates $\varnothing 50\text{mm} \times 10\text{mm}$ were ground flat and parallel $<1\mu\text{m}$ and spin-coated with a precursor layer. After heating to 400°C for 1 hour, a nano-crystalline inorganic ZrO_2 layer was formed with a thickness between 100nm and 150nm . This process had to be repeated several times, in order to achieve a total layer thickness of at least $1.5\mu\text{m}$. Crack free deposition of such multi-layer ZrO_2 coatings was possible up to a thickness of $2\mu\text{m}$ [2].

Face turning experiments with single crystal diamond tools were carried out with a nominal depth of cut of $1\mu\text{m}$ and a feed of $2\mu\text{m}/\text{rev}$. The average cutting speed was 1.2 m/s . Diamond tools with a nose radius of 3mm and rake angles of 0° and -45° were used. After a cutting distance of approx. 2000m both tools showed excessive rounding of the cutting edge (cf. Fig.1, left). The diamond turned surface was covered by randomly distributed cracks (cf. Fig.1, right). Formation of cracks as well as tool wear was found to be independent of cutting speed and depth of cut.

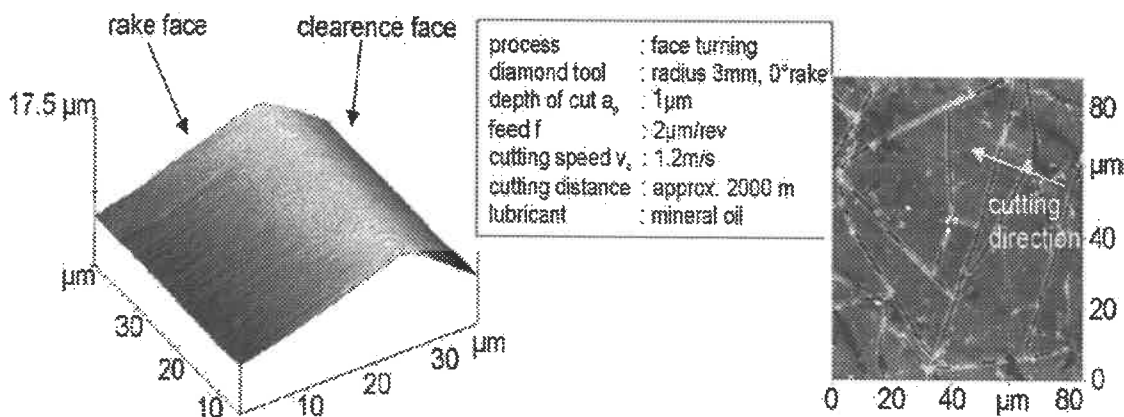


Figure 1: Scanning force microscope images of cutting edge after diamond turning of sol-gel derived ZrO_2 layer (left) and of the diamond turned surface (right)

Ormosil coatings

Ormosil coatings contain an inorganic and an organic compound and are somewhat softer than sol-gel derived ceramic coatings. ZrO_2 ormosil coatings were produced by spin-coating a comparatively thick precursor layer ($20\mu\text{m}$) and heating to 100°C for 24 hours.

Face turning experiments were carried out with diamond tools with a nose radius of 3mm and rake angles of 0° , -10° and -20° , resp. Diamond turning of ZrO_2 ormosil layers did not lead to any detectable tool wear over a cutting distance of at least 4000m . Moreover, crack free surfaces could be generated. A significant dependence of surface roughness on cutting speed could not be observed. However, surface roughness strongly depends on the rake angle of the cutting tool. The results of 125 individual roughness measurements carried out with a white-light interferometer are shown in Fig 2.

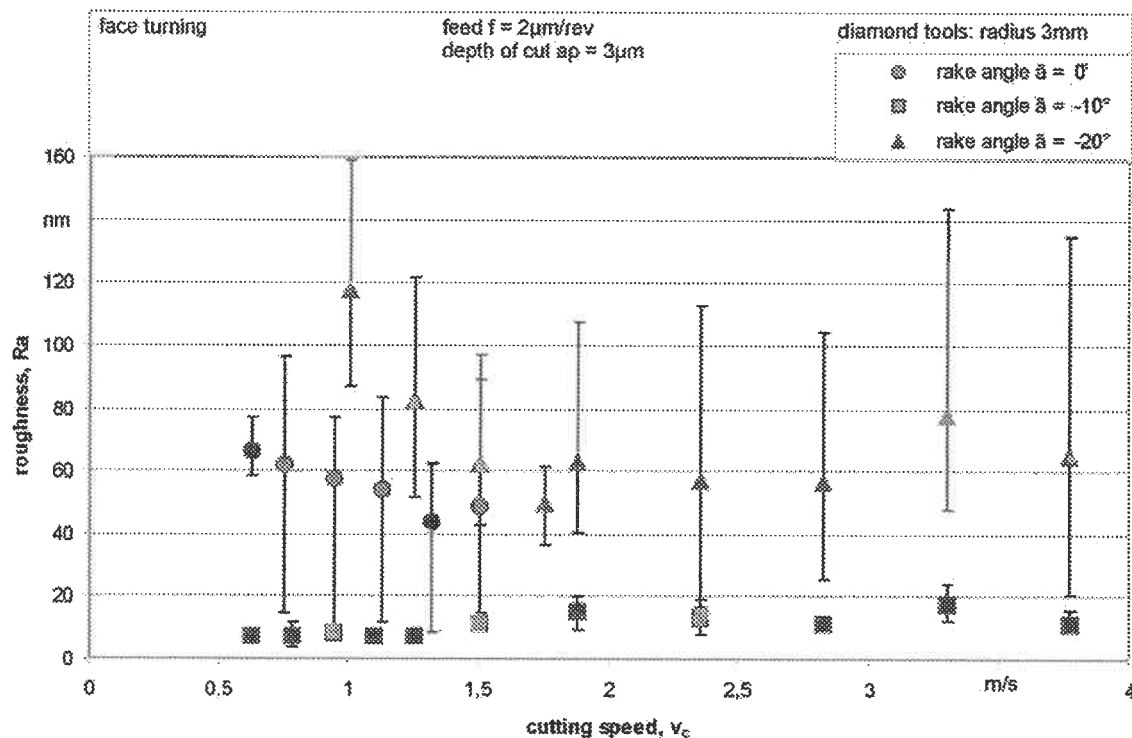


Figure 3: Ra surface roughness of diamond turned ZrO_2 ormosil coatings vs. cutting speed

Since ZrO_2 ormosil coatings can be deposited with a thickness up to $20\mu\text{m}$, they bear the potential of being microstructured by diamond machining. We have carried out fly-cutting experiments with a pointed diamond knife with an included angle of 70° . Two groove systems intersecting under 90° exhibit very little burr formation at edges and corners, but also reveal occasional delamination at corners (cf. Fig.3). Hence, adhesion of ormosil layers to the substrate has to be improved.

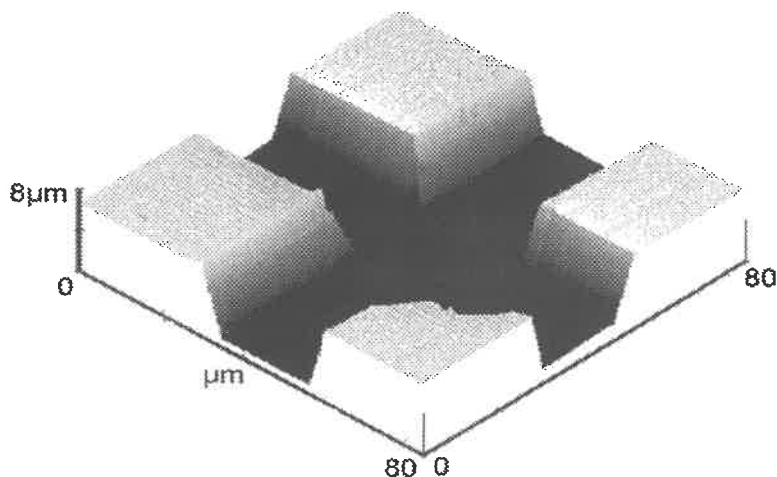


Figure 2: Scanning force microscope image of intersecting V-grooves fly-cut into a ZrO_2 ormosil layer

TiN and TiCuN PVD-coatings

The stoichiometric TiN and TiCuN PVD-coatings were provided by the Institute for Materials Science (IWT), Bremen. The coatings were deposited on S-6-5-2 steel substrates ($\varnothing$ 40mm x 10mm) with a thickness of approx. 20 μ m [3;4].

Peripheral as well as contour grinding experiments were carried out with three different types of diamond grinding wheels: a resin bonded fine grained, a metal bonded fine grained and a metal bonded coarse grained wheels (cf. Table 1).

bond material	Concentration	average grain size	diameter
resin	C75	2 μ m	75mm
metal (cast iron)	C75	3 μ m	100mm
metal (electroplated)	-	91 μ m, truncated	120mm

Table 1: Applied grinding wheels for peripheral and contour grinding experiments

A special dressing technology was applied to the coarse grained metal bonded wheel for truncating the tips of the grains yielding a special wheel topography consisting of flattened grains with negative rake angles within a narrow envelope. Such grinding wheels had already been applied successfully for ductile grinding of optical glasses [5].

A surface roughness between 20nm and 50nm was obtained with the fine grained resin bonded wheel both in peripheral and contour grinding modes (cf. Fig. 4). Both TiN and TiCuN showed a ductile material removal behaviour. Predominately brittle material removal was observed with the metal bonded wheels. No better surface finish than Ra 80nm could be achieved.

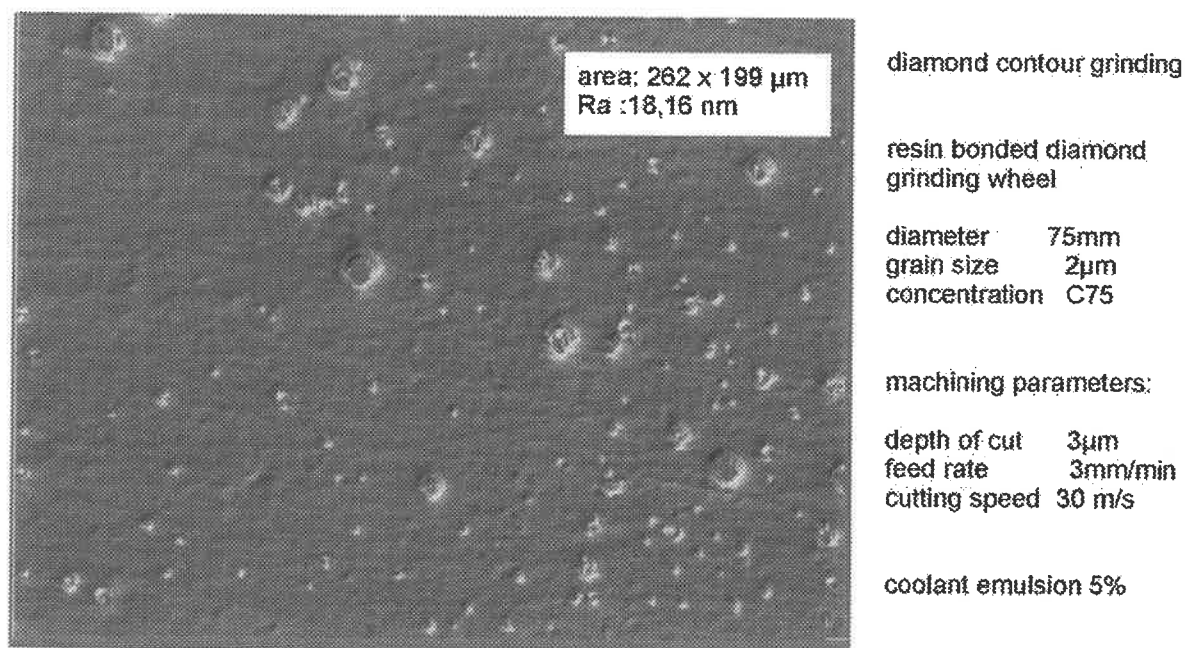


Figure 4: White-light interferometer image of a stoichiometric TiN coating ground in ductile mode with a resin bonded fine grained wheel

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Temperature Effects in Micro-machinability of Plastics

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1. Introduction

Plastics are very useful material as their specific gravity are small and as we can design and adjust desirable mechanical, optical or chemical property of the material to some extent. When you intend to use the plastics to micro parts or fine parts, high accuracy or small surface roughness is needed. Usually plastic parts are made by some molding technology like an injection molding. However the technology has some difficulty to attain preciseness because of shrinking during cooling process. Thus in order to get a ultra-precise plastic parts you should apply a machining process, for example diamond turning or grinding, with a very small working unit (bit size). But there are a lot of kinds of plastics and it is thought that attainable working unit depends on the kind of plastics. However machinability, especially micro-machinability of the plastics has not been revealed yet. In our previous study we found the surface roughness greatly differs by the sample materials. In amorphous plastics like PMMA, PC or PS, the surface roughness is rather good and becomes worse with the increase in the feed per revolution. In this study we focused the temperature effect of micro-machinability of plastics as most plastics are more temperature sensitive than metals or ceramics. This study is concerned with the question that how the temperature generated by cutting operation effects to the micro-machinability of plastic materials.

Table 1 Experimental condition

Specimen plastics	Crystalline	Polyamide 6 (PA6) Polypropylene (PP) High density polyethylene (HDPE)
	Amorphous	Polycarbonate (PC) Polyvinyl chloride (PVC) Polystyrene (PS) Polymethyl methacrylate (PMMA)
Size of specimen		20mm(diameter), 1mm(thickness)
Cutting method		Fly cutting
Tool		Single crystal diamond (nose radius=5mm)
Cutting speed		600m/min
Feed		2-20 micrometer/rev
Depth of cut		2-20 micrometer

2. Experiments

Cutting tests with round nose diamond tool were conducted. The experimental condition is shown in Table 1. Seven kinds of plastics also shown in Table 1 were evaluated. The temperature at the center on the rear surface of each specimen was measured by a thermocouple during the cutting test.

The experimental results of specimens when

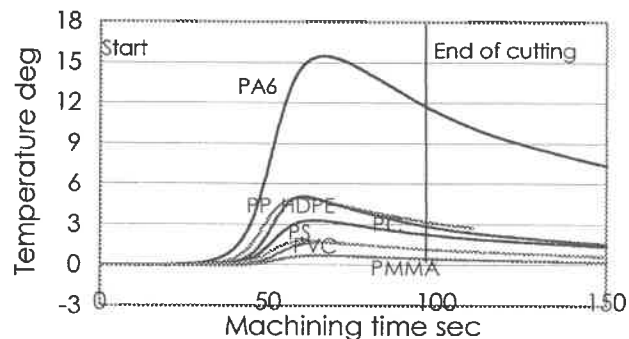


Fig.1 Temperature transition during surface cutting

setting the feed per revolution to 5 micrometers and the depth of cut to 5 micrometers are shown in Fig.1. The time zero means a start of the cutting and cutting one surface completed for 100 seconds. Even in the same cutting condition, the crystalline plastics make the temperature on the rear surface raise higher than that on the amorphous plastics. The temperature rise on PA6 was the highest, and that on PMMA was the smallest.

Fig.2 shows the experimental results about the relation between the feed rate and the temperature rise in each material. The depth of cut was fixed to 5 micrometers. An amount of the temperature rise was the highest temperature in each cutting test (see Fig.1). Along with growth in a feed per revolution, the tendency for the temperature rise of a specimen to decrease was acquired. Moreover it also shows that the crystalline plastics make the temperature raises higher than that on the amorphous plastics.

3. Simulation and discussion

3.1. Inflow heat flux by cutting operation and cooling effect

In the next step we conduct a three dimensional heat conduction simulation by finite differential method. Parameters of the simulation were an inflow heat flux to the surface by the cutting operation and a heat transfer from the surface by the surrounding air (the cooling effect). Mesh size was 0.5mm on surface (40 divisions by 40 divisions to 20 mm diameter) and 0.1mm along depth (10 divisions to 1mm thickness).

To a specific experimental condition we calculated a lot of times by changing the inflow heat flux or the heat transfer from the surface in order to find out a temperature transition which is similar to the experimental data. Fig.3 shows an example for PA6 in which the feed rate was 5 micrometer per revolution and the depth of cut was 5 micrometer. We judged the experimental result and the simulation

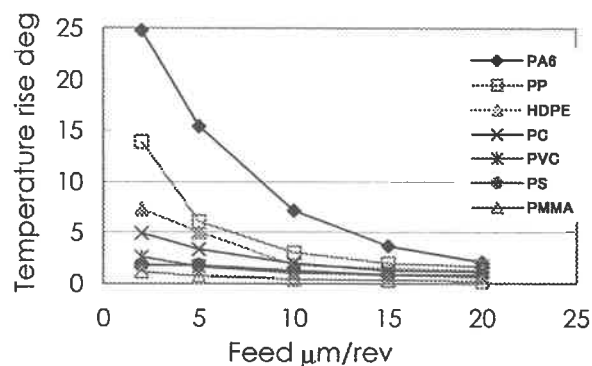


Fig.2 Relation between feed and temperature rise in various plastics

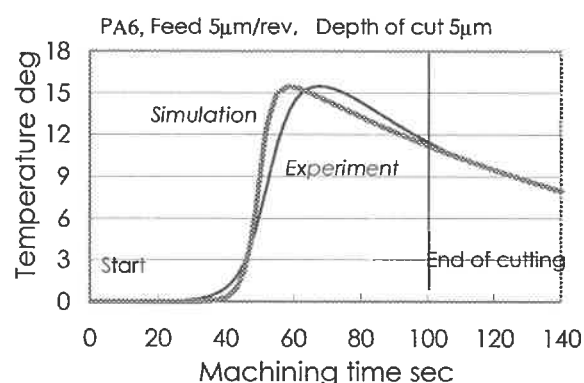


Fig.3 Temperature transition on the rear surface by the experiment and the simulation

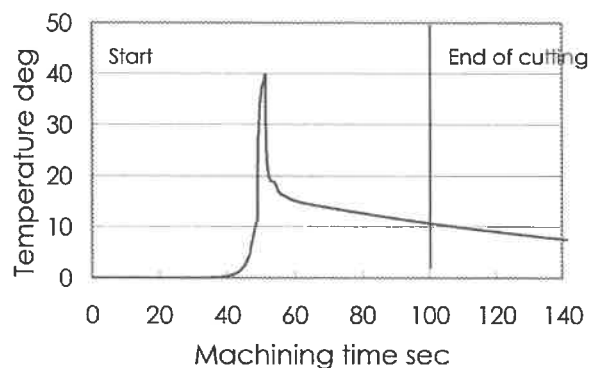


Fig.4 Temperature transition on the cutting surface calculated by the simulation

coincided. By using the amount of the inflow heat flux and the heat transfer coefficient which were used in the simulation, we could estimate these parameters.

The calculated temperature transition at the center on the cutting surface of PA6 is shown in Fig.4. It shows that the temperature rise on the cutting surface is very sensitive to tool's passing and that is also faster and higher than the temperature on the rear surface by comparing with Fig.3.

3.2. Surface temperature on a specimen and glass transition temperature of each material

By the heat conduction simulation, the inflow heat fluxes by cutting operations were guessed. The amounts of inflow heat flux in various materials and with various feed rate are shown in Fig.5. The heat generated in cutting operation is thought to consist of a deformation resistance (energy) of material, a friction between a rake face and chips, and a friction between a flank face and a specimen. If the feed rate increases, the deformation energy should increase but the frictions may decrease because total contacting time between tool and workpiece decreases. As a result, there is not any tendency between the feed rate and the heat influx in Fig.5.

Fig.6 shows the simulated temperature on the cutting surface in each material under the condition which the feed rate was 2 micrometer per revolution and the depth of cut was 5 micrometer. A horizontal axis of Fig.6 is a glass transition temperature of each material. The cutting temperature of HDPE and PP is much higher than their glass transition temperature, and the cutting temperature of PA6 is almost the same as its glass transition temperature. Moreover, it become clear that the cutting temperature of amorphous plastics is quite lower than their glass transition temperature. In PP or HDPE case, it is imagined that their machined surface is worse as the mechanical or physical properties of the specimens change a lot (become softer) during the cutting operation.

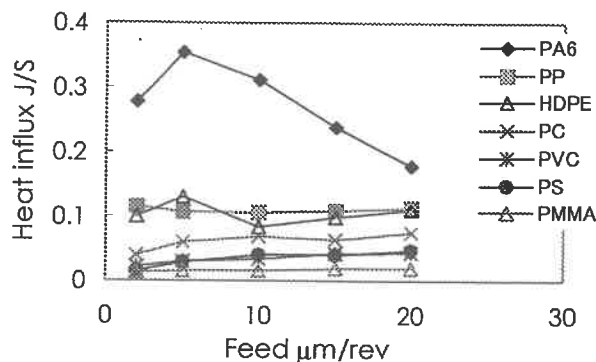


Fig.5 Heat influx by cutting operation in various material and various feed rate

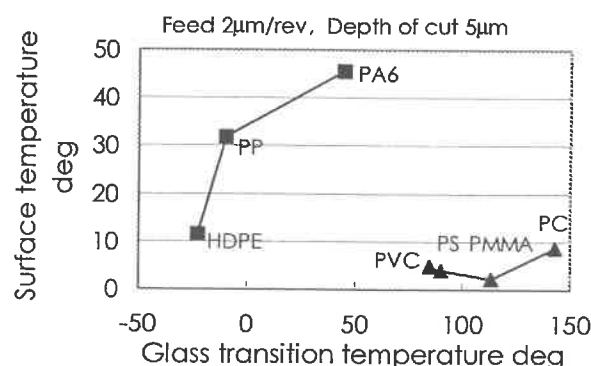


Fig.6 Cutting temperature and glass transition temperature in each material

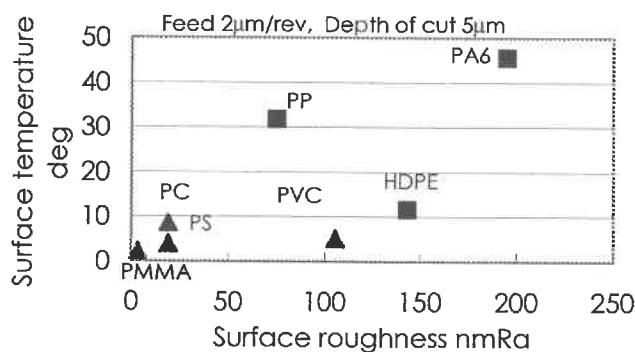


Fig.7 Cutting temperature and surface roughness in each material

The temperature on the cutting surface under the condition of the feed of 2 micrometer and the depth of cut of 5 micrometer in each material is shown in Fig.7. The horizontal axis of the Fig.7 is the surface roughness of the specimen with the same machining condition. Even in the same cutting condition, there is a slight tendency for the surface roughness to become worse with higher surface temperature from the higher inflow heat flux.

4. Conclusion

Cutting temperature measurement and the heat conduction simulation of seven kinds of plastics were performed. The presumption of the cutting surface inflow heat flux by cutting operation and the cutting surface temperature were completed. Comparing the cutting temperature with the glass transition temperature of each material, we derived some conclusions from this study. Those are as follows.

- (1) The temperature of the surface decreases with increasing of the feed rate. The temperature increases with increasing of the depth of cut.
- (2) In most of non-crystalline plastics the surface roughness is good as their surface temperature is much lower than the glass transition temperature.
- (3) In crystalline plastics the surface roughness is bigger as the surface temperature is higher than the glass transition temperature.
- (4) If the cutting temperature is high, the surface roughness becomes worse.

Study on Slurry Actions in Slicing Groove and Slicing Characteristics at Multi-Wire Saw

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1. Introduction

The multi-wire saw of the workpiece descent-type is one of the methods to slice the large-sized silicon ingot. This slicing method uses the tensioned thin wire and the working fluid mixed abrasive grains, which is called such as slurry. In case of the multi-wire saw of the workpiece descent-type, this slurry is supplied on the wire and carried in the processing area. The workpiece is processed by slurry that enters between the wire and the workpiece. Although the slurry actions and the slurry compositions are very important to slicing efficiency and accuracy of silicon wafers, there are few papers that treat the relation between the slurry actions and slicing characteristics. Therefore, the supplying method of slurry is selected empirically and by means of know-how of the worker or methods and conditions recommended by the manufacturers of multi-wire saw. Authors have been paid their attentions to the slurry actions around the workpiece in case of the multi-wire saw of the work descent-type [1], [2].

In this paper, the slurry actions at the periphery of the wire and the inside of the slicing groove were observed by a high-speed video camera. The slicing characteristics were studied under a series of slicing conditions by using the multi-wire saw of the workpiece descent-type. And, the relations between the slurry actions and the slicing characteristics (the slicing efficiency and the slicing accuracy) were clear.

2. Observation methods of slurry actions

Figure 1 shows the model of slurry supplying method at the multi-wire saw of the workpiece descent-type. As shown in Figure

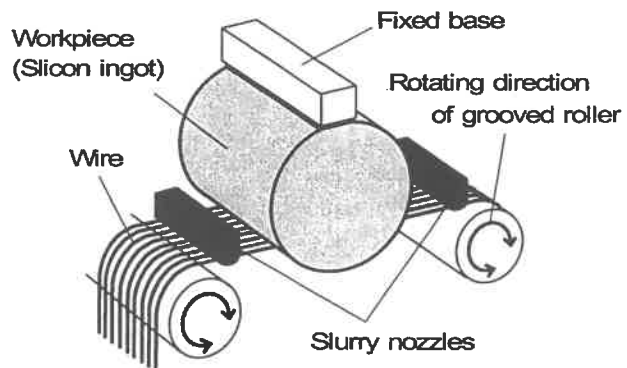


Fig. 1 Model of slurry supply method

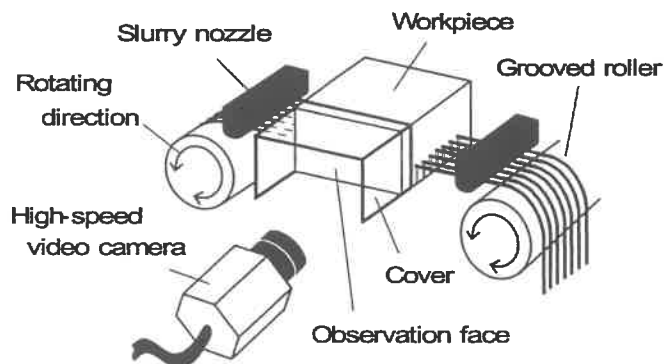


Fig. 2 Observation of slurry actions in groove

1, the slurry is supplied on the wire near the workpiece and carried into processing area.

Figure 2 shows the observation model of slurry actions in the slicing grooves. The observation of slurry actions in the grooves was carried out by means of the rectangle type of soda glass, and the observed face of the soda glass was buffed.

Table 1 shows the main slicing conditions.

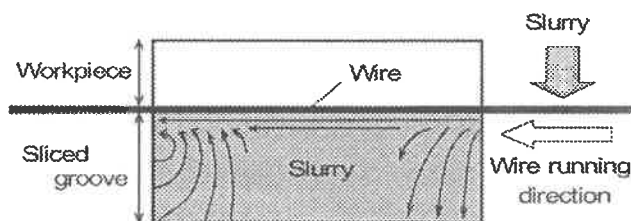


Fig. 3 Model of slurry flow in slicing groove

3. Slurry actions in the slicing grooves

Figure 3 shows the model of slurry flow in the slicing grooves, and the arrow signs mean the slurry flow in the grooves. As the results of this observation, the whole of slicing grooves was filled up with slurry. It is clear that the slurry in the slicing grooves moves to below side at the entering area of wire and moves to wire side from side and bottom surface at the leaving (exit) area of wire.

Figure 4 shows the observation photos in the slicing grooves. The multi-wire saw used in this experiment is a type of reciprocating wire motion. The photos from No.1 to No.4 in Figure 4 show the slurry behavior at a series of wire motion (between starting time of wire running and stopping time to reverse the wire running). As shown in Figure 4-(a)-No.1, there are some bubbles of various size in the slicing grooves. These bubbles form air layer

Table 1 Main experimental conditions

Workpiece (size)	Soda glass ($60^W \times 50^D \times 10^H$)
Slurry (Concentration)	PS-LW-1 + GC#800 (35wt%)
Wire, Wire diameter	Piano wire, $\phi 0.2$
Wire running speed	0~400 m/min
Feed speed of new wire	2.0 m/min
Wire tension	19.6 N
Wire pitch	2.5 mm
Working load	0.65 N/wire

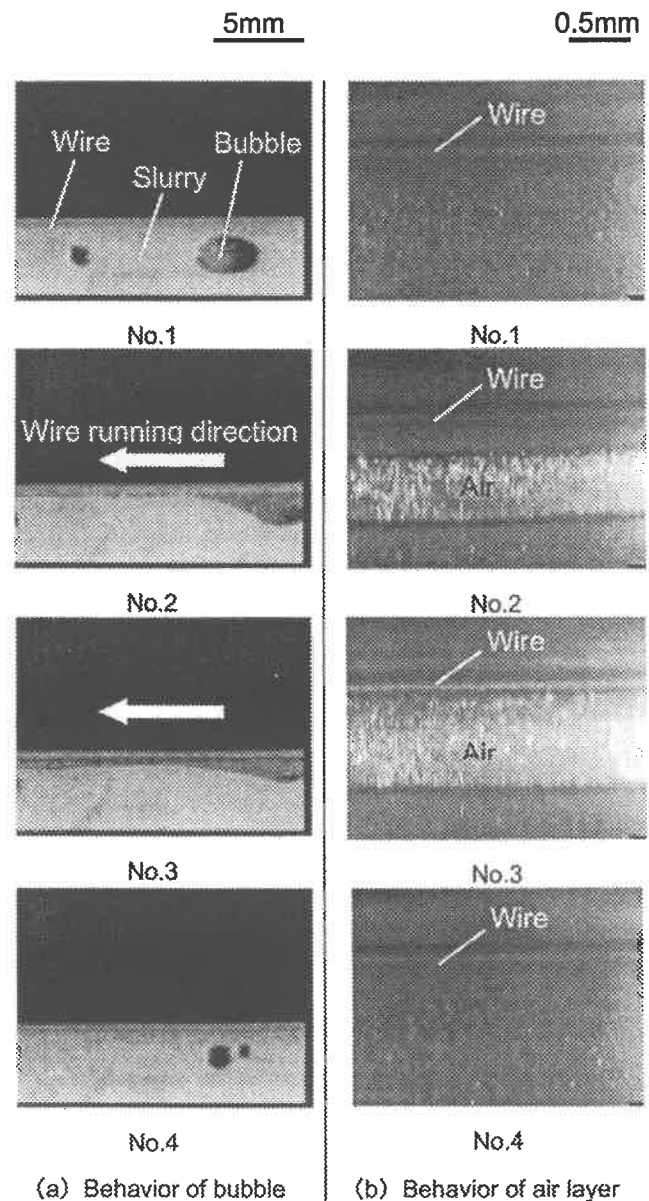


Fig. 4 Observations in slicing grooves
(Center of workpiece)

around the wire due to wire running (see photos of No.2 and 3), and the air layer returns to bubbles at the stop of wire running. Figure 4-(b) shows the results observed around the wire. As shown in Figure 4-(b), it is clear that the air layer prevents slurry from entering between wire and workpiece.

4. Entering mechanism of bubbles in the slicing grooves

Figure 5 shows two types of slurry actions at the supply point of slurry. One is the condition that sticks the slurry on wires (see Figure5-(a)), and another one is the condition that films with slurry among the wires (see Figure5-(b)). Authors defined these conditions such as no-film and horizontal films.

Figure 6 shows air actions in the slicing groove around entrance area of wire. The air enters to slicing grooves from the side surface of workpiece with wire running.

Figure 7 shows air actions in the slicing groove around center and exit area of wire. Airs enter to slicing grooves from side and bottom surface of workpiece with wire running. It is clear that much air enters in the slicing grooves through observation in case of the slurry supplying method used in the multi-wire saw of the workpiece descent-type.

5. Relationship between slurry actions and processing characteristics

Figure 8 shows the behavior of slicing efficiency in case of no-film and horizontal films. The slicing efficiency in the condition of horizontal films increases to be

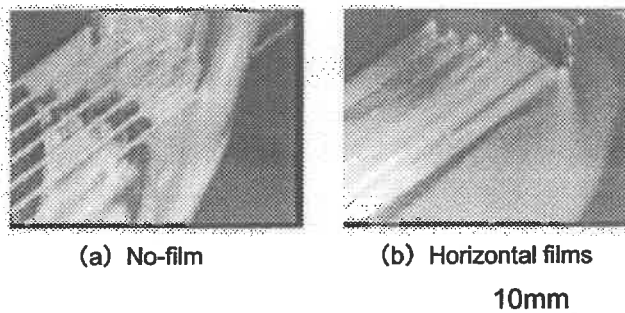


Fig.5 Behavior of slurry actions at supply point

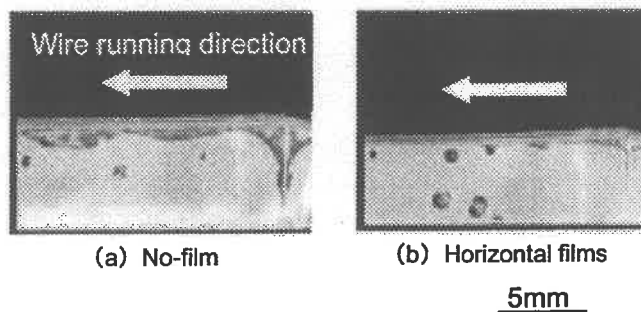


Fig.6 Air actions in slicing groove at entrance area of wire

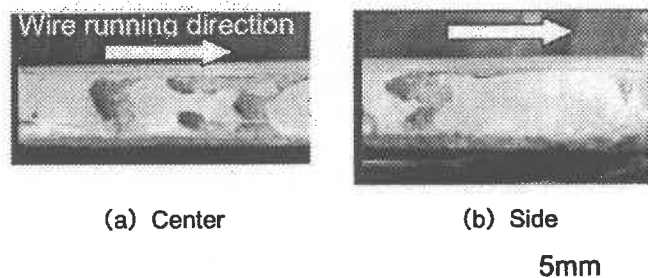


Fig.7 Air actions in slicing groove at center and exit area of wire

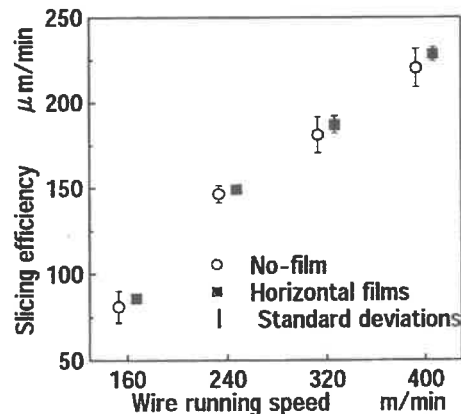


Fig.8 Behavior of slicing efficiency

compared with the condition of no-film, and the standard deviations in the horizontal films become small value. It is clear that the slicing in the condition of horizontal films is better than no-films slicing.

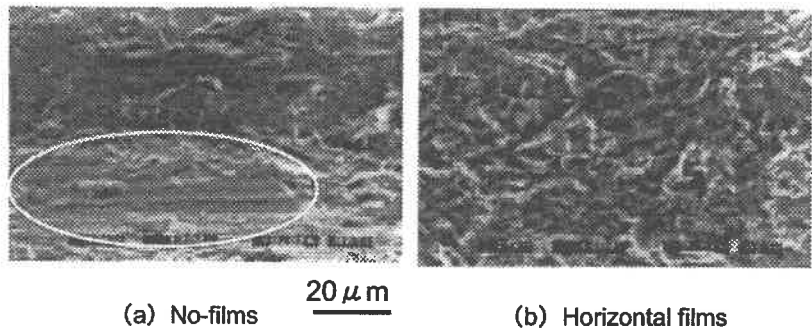


Figure 9 shows the SEM observation photos of apex surface of slicing groove. These photos show the observation results of the center points in the slicing grooves. As shown in Figure 9-(a), the scratch caused by contact between the wire and workpiece is observed. This phenomenon means to be brought about by the shortage of slurry due to air layer around the running wire. On the other hand, the slicing grooves in the condition of horizontal films form uniform satin surface. Figure 10 shows slurry actions in case of the dipping method. When the dipping method is used for the slurry supplying method of multi-wire saw, slurry in the slicing grooves has no-bubbles.

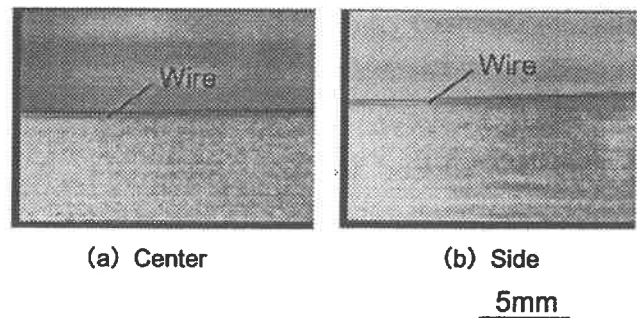


Fig.10 Slurry actions in slicing groove in case of dipping method

6. Conclusions

Slurry actions at the multi-wire saw of the workpiece descent-type are studied, and the followings are clear.

1. The slicing grooves are filled by slurry and some bubbles, and the wire running during the processing brings about the slurry flow in the slicing grooves. These bubbles in the slicing grooves join by this slurry flow and the bubbles become air layer around the wire.
2. When the air layer appears around the wire, the slurry is not supplied into the processing area between the wire and the workpiece. And, these phenomena become factors of low slicing efficiency and low accuracy.
3. The slurry can enter in the slicing grooves from the both side surface of workpiece and the blow surface of workpiece due to the slurry flow in the slicing grooves. But, small amount of the slurry is supplied in the slicing grooves in case of the normal slurry supplying method used at the multi-wire saw of the workpiece descent-type, and the phenomena of entered air in the slicing grooves bring about the shortage of slurry.

The references are omitted.

Effects of Nitrogen Gas Atmosphere in Oxygen-free Copper Precision Cutting

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key words: Nitrogen gas atmosphere, oxygen-free copper, precision cutting, diamond turning, oxygen gas, tool wear, redox, Linear Collider, elementary particle physics

1. Introduction

In elementary particle physics, an important goal is to build a practical Linear Collider providing electron beams with over 100 Giga electron volts [1]. To investigate the high energy elementary particle reactions, the Linear Collider would accelerate electrons and positrons towards each other and the physicist would then observe the elementary particles that are generated at the collision point [2]. The Linear Collider requires a number of ultraprecision-machined accelerator cells of oxygen-free copper that have an excessive number of configurations [3]. In the mass production of the cells, a variety of diamond tools would be consumed, and so it is necessary to improve the life of the diamond tool from a financial viewpoint. However, cutting fluids cannot be used during the diamond tool cutting as cutting fluid particles accumulated on the diffusion-bonded cell surfaces [4]. Recent research has reported that a redox reaction seems to occur between the copper atoms and the carbon atoms during the ultraprecision oxygen-free copper cutting by the diamond tool [5]. In this redox reaction, the carbon atoms of the diamond tools are oxidized to CO or CO₂ by CuO and the reduction in the tool life is result of developing micro cracks that are generated during the tool grinding. It is also reported that the embrittlement of the diamond tool could be prevented by inhibiting the redox reaction with a N₂ gas atmosphere around the cutting part [6]. However, these have only been preliminary experiments, and there has been little research that clearly describe the effects of the N₂ gas atmosphere on the ultraprecision machined surface and the diamond tool life. In this paper, the effects of the N₂ gas atmosphere on precision cutting phenomena have been investigated.

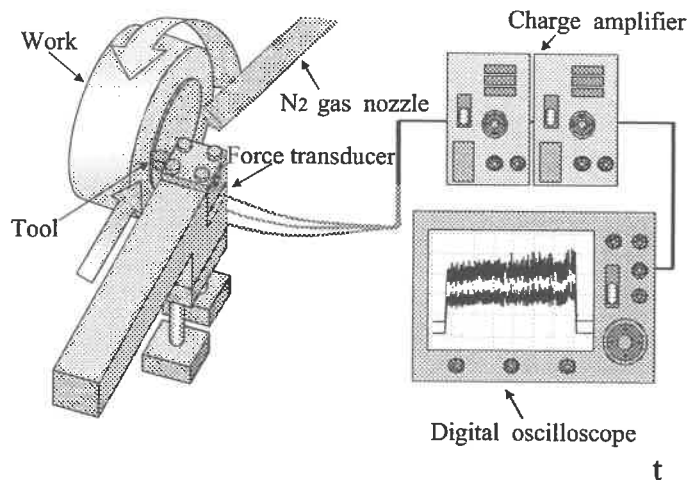


Fig.1 Experimental setup for the gas atmosphere effects on the cutting

Table 1 Cutting conditions

Work material	Oxygen Free Copper, 62 •															
Tool material	Cemented carbide K10, 1 P M Z D S Z T H U B M M J O F E J B N															
Tool geometry	Rake angle	0														
	Relief angle	7														
	/ P T F	E O H M F														
Cutting form	Surfacing															
Cutting speeds V m/min	92 253 (Constant 1300 rpm at main spindle)															
Depth of cut t , m	20															
Feed f m																
Cutting fluids	Dry, N ₂ gas, O ₂ gas															

2.Experimental procedure

An CNC lathe was conducted on the gas atmosphere effect cutting experiment.

The work material was vacuum-annealed oxygen free copper cylindrical bars (99.99% purity) having a 62mm diameter.

The tools used were tungsten carbide tools K10 and polycrystalline diamond tool (PCD, Allied Material Co.) which have 135° nose, 0° rake, and 7° clearance angle.

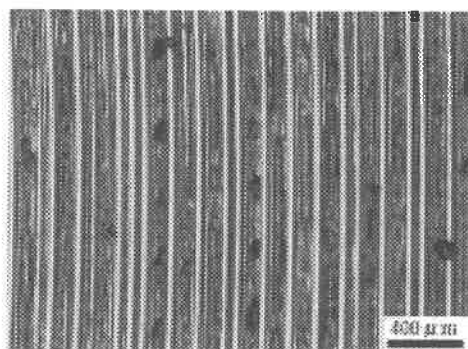
To investigate the effects of the Nitrogen (N₂) gas atmosphere, the gas was supplied to the cutting part at a speed of 55.6 m/s. Then, the partial pressure of oxygen was detected. The pressure around the tool edge was 0.014Mpa. Oxygen (O₂) gas was also used to investigate gas blow effect, that is to say, aerodynamic effect on the chip flow. Fig.1 shows general view of the setup of gas atmosphere effect cutting.

A quartz force transducer (Kistler Co. type 9251A) was employed to measure the cutting forces.

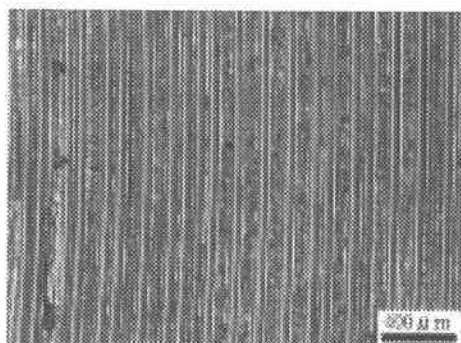
Feed was 10, 50 mm for the PCD, cemented carbide tool cutting respectively. The preliminary cutting tests indicated that those values in the feed can produce the best quality-machined surface

Table 2 Gas atmosphere effects on surface roughness and cutting forces

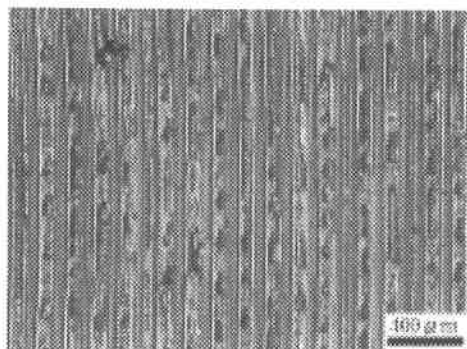
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\$ V U U F O H	G M V % S Z					H B T H	B T % S Z				A B T H B T	/									0
4 V S G B D F3	S P V H I O F																				
\$ V U U F O	H G P S D																				
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a) Dry



b) N₂ gas



c) O₂ gas

Fig.2 Micrographs of machined surfaces by cemented carbide tool

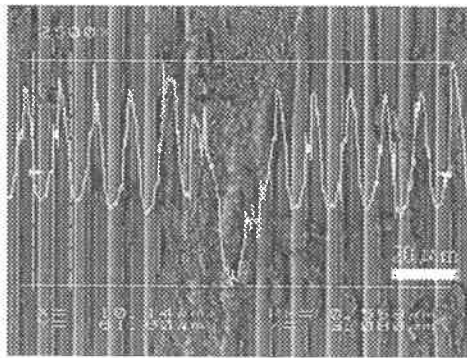
in each tool cutting. After completing the experiments, the machined surface was observed, and its surface profile was measured.

Table 1 summarizes the cutting conditions.

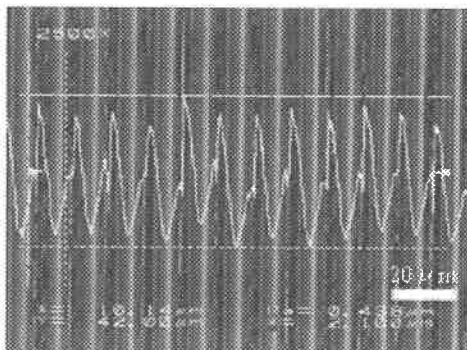
3.Experimental results

Table 2 shows the surface roughness and the cutting forces in the three types of gas blow cutting with the two kinds of tools. The cutting forces F_T and F_N in the table means tangential, normal component, respectively. The cutting forces in N₂ gas offered the lowest values, and slightly lower than in O₂ gas. As for fluctuation in the cutting forces, the values in the dry fluctuated to a great extent more than in N₂ and O₂ gas. It suggests chip formation in the dry was unstable.

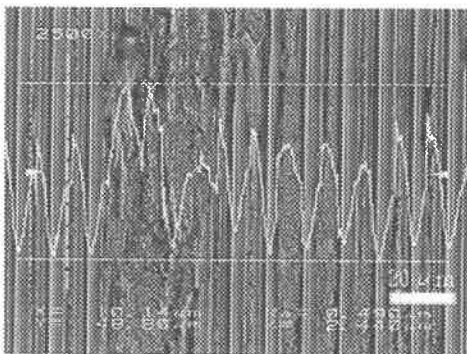
Fig.2 shows optical micrographs of the machined surfaces by the cemented carbide tool. In the dry cutting, at first glance the feed marks could be clearly found, however, a closer examination showed that some of the cutting grooves were not recognized, that is to say, side flow perpendicular to the cutting direction covered the grooves in places. The size of the side flow can be controlled by properties of work hardening and ductility of the work material and friction between the tool rake face and the chip. Some residues, shown as black spots in the figure, could be recognized. On the other hand, the surface finish in the N₂ gas was smooth better than in O₂ gas.



a) Dry



b) N₂ gas



c) O₂ gas

Fig.3 Micrographs of machined surfaces by polycrystalline diamond tool

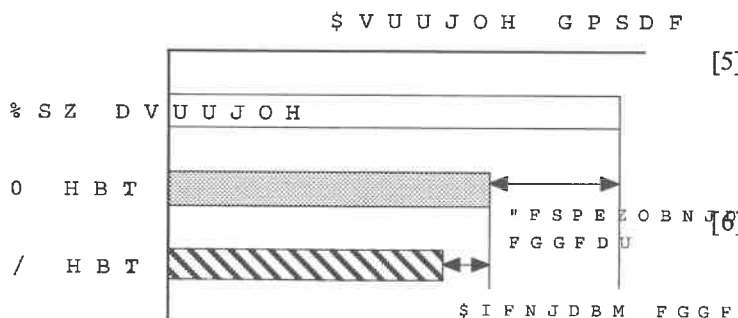


Fig.4 Idea of reduction in the forces

Fig.3 shows the machined surfaces by the PCD. The lines on the image were surface profile curves measured by an optical profile meter. The surface finish and the cutting forces by the PCD presented the same trend as by the cemented tool. The surface finish in the N₂ gas was superior to the others. As for ability to transfer the tool shape to the machined surface profile, namely transference, the cutting in the N₂ gas offered the best performance. In addition, the cutting forces was the smallest among them. Consequently, it was determined that the N₂ gas atmosphere improved the surface finish and the cutting forces in the oxygen-free copper cuttings more than the others.

Fig. 4 shows general idea for reduction components in the cutting forces. O₂ gas appears to make an aerodynamic effect which tear off the chip from the work material. Furthermore, N₂ gas seems to make chemical reaction at the interface between the chip and the work material.

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Application of the run length encoding algorithm to the fabrication of non-rotationally symmetric surfaces

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Abstract

When a non-rotationally symmetric surface with complex shape is machined in diamond turning machine, Fast Tool Servo(FTS) can be an effective means of fabricating non-rotationally symmetric components. But the size of geometric data describing the surface becomes very large because the depth of cut varies in θ direction as the spindle rotates. Since the FTS controller has a limited capacity of memory, it is very difficult to transfer the data file at one time. The method of splitting the data file into several ones and transferring them to the machine controller by rotary buffer, was undesirable when we have a small amount of memory and there is no more remaining cycle time in FTS controller. In this paper, we propose the method of compressing the large size geometric data file into the smaller ones by run length encoding algorithm. Also, through this algorithm, we can recover the original data file by decompressing the encoded file.

Key words : Run length encoding, Fast Tool Servo, non-rotationally symmetric surfaces

1. Introduction

Due to the low bandwidth of Z-axis in the diamond turning machine, FTS is needed for the fabrication of non-rotationally symmetric surface with complex shape. We implemented FTS which is actuated by piezoelectric actuator incorporated with capacitive displacement sensor and has a high bandwidth of following the fast motion resulting from the variant motion of tool tip in non-rotationally symmetric surfaces. The FTS has a sufficiently high stiffness for the diamond turning process and it is shown that the FTS can be an effective means for the real-time error correction in the machining of flat surfaces when the machine has a movement error resulting from the spindle growth[1]. Also, if we apply the fast actuating capability of FTS into the fabrication of non-rotationally symmetric surfaces, the complex shape can be easily machined in T-type diamond turning machine [2].

In this paper, we propose the method of compressing the large size geometric data file into the smaller ones by run length encoding algorithm. This method has been originally used in computer vision technique for the purpose of reduction of size of image data files. It is known that when the same number in data file occurs repeatedly in the contents, this method is very effective. Therefore, the run length encoding method can be used effectively for the reduction of data size because of the depth of cut of FTS is occasionally repeated in non-rotationally symmetric machining.

2. Principle and Structure

In non-rotationally symmetric machining, the surface profile has the components varying in θ direction as the spindle rotates. And the conventional Z-axis in diamond turning machine cannot move fast enough to

First, in the case of run length encoding, when we are given the original data array $P(n)$ which is to be encoded, we should decide how many repeated points the data array has in its sequence. Therefore, we introduce the temporary variable Tmp and compare it with the original data array $P(n)$ increasing the index n . Because the temporary variable Tmp is set to the $P(n-1)$ in the $n-1$ time step, the encoded data array $Q(m)$ should increase until the Tmp is not equal to $P(n)$ in n time step. And when the Tmp is equal to $P(n)$, $Q(m)$ should stop increasing and set to be 0. When the index n is greater than N , the total number of original data array $P(n)$, the iteration process is finished.

On the contrary, in the case of run length decoding, it should be determined whether the index m is odd or not when there is an data array $Q(m)$ which is to be decoded. If m is odd, the 1's are occurring in data file repeatedly corresponding to the value of $Q(m)$. And if m is even, the 0's are filled up in the sequences of an interval of next $Q(m)$ points. This process is repeated until the index m is greater than the number of data array M .

3. Experiment

For the simulation of run length encoding method, the machining data array for alphabet letter 'A' was generated as shown in Fig. 3. The spindle speed was set to 1000 rpm and feed rate was 100 mm/min.

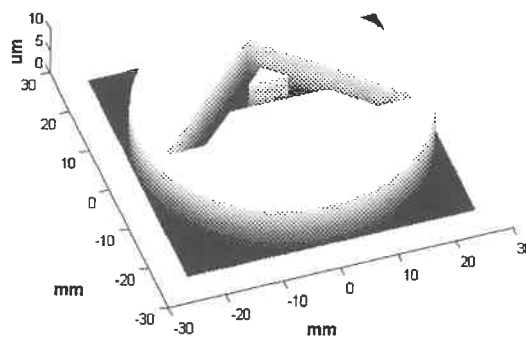


Fig. 3 The 3-D shape of 'A' alphabet for simulation

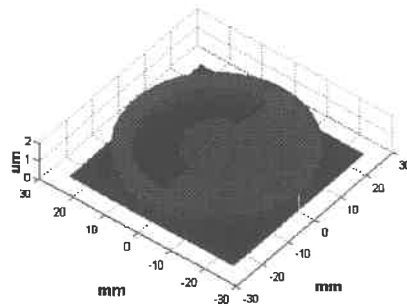
The width of letter 'A' is 44 mm and length is 15 mm. The number of original data array is 500,000. This number of data cannot be handled in FTS controller because of the limited memory. Through the proposed run length encoding algorithm, the original data array is compressed and the number of data is reduced from 500,000 to 403. For the verification of proposed method, we decompressed the compressed file by run length decoding algorithm and its 3-D shape is shown in Fig. 3.

For the machining experiment, the face cutting for the generation of the simplest non-rotationally symmetric surface was carried out. As in Fig. 4(a), this simplest non-rotationally symmetric surface has just 2 types of depth of cuts, 0 and 1, which vary twice per spindle rotation. The FTS was programmed to move forth $3.4 \mu\text{m}$ for the depth of cut 1 and return to the zero position in the case of depth of cut 0. Therefore, the machined workpiece has a shape like a step function along the θ direction and its period is one revolution. The photograph of machined surface is shown in Fig. 4(b). It was machined at 150 rpm and the feed rate was set to 1.5 mm/rev. For the optical quality of workpiece, the machining experiment was performed in commercial diamond turning machine along with FTS. Although this machine is being used for mass production of the optical lens, the spindle encoder cannot be connected to the counter interface of the FTS controller. Therefore, the relations between the encoder pulse numbers and spindle rpm was used for the generation of non-rotationally symmetric surface [2] as in Equation 1.

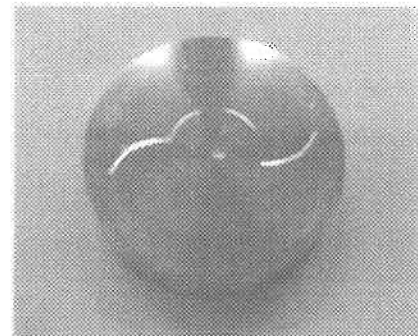
$$\Delta t = \frac{1}{E \times RPM} \quad (1)$$

where Δt is the time per one pulse and E is the number of encoder pulses per one revolution and RPM is the number of revolution per one minute. The size of the original data file was 39 MB and we can obtain the compressed file with its size of 41 KB by run length encoding algorithm.

The machined result shows that FTS can fabricate the simplest non-rotationally symmetric shape as shown in Fig. 4(b) but the line which divides the depth of cuts 0 and 1 along the θ direction is not perfectly straight in Fig.4(b). This is due to the fluctuations of the spindle speed in the commercial diamond turning machine, indicating that FTS was not perfectly synchronized to the spindle encoder. If we can use the encoder pulse as an input of the run length encoding method, it is expected that more precise machining can be possible.



(a) The 3D profile of the simplest type of surfaces



(b) The photograph of machined surface(Al 6061)

Fig.4 The simplest type of non-rotationally symmetric surfaces

4. Conclusion

By run length encoding algorithm, the geometric data which is composed of the values of depth of cut in θ direction, has been successfully compressed and inputted to the FTS controller at one time. For the generation of the surface profile data, the compressed file was decoded and transformed into the form FTS can handle effectively. Since FTS controller receives the whole compressed file at one time before the machining, its memory size does not have to be large. Therefore, the non-rotationally symmetric surface which requires the large amount of data can be fabricated more effectively with run length encoding algorithm.

Acknowledgements

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Nanostructured Yttrium-Stabilized Zirconia/Alumina (YSZA) Cutting Insert for Machining High-Speed M1 Tool Steel

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Abstract

This research studies the performance of cutting inserts made of nanostructured yttrium-stabilized zirconia/alumina (*n*-YSZA) in machining of high speed M1 tool steel. This paper briefly describes the fabrication of the nanostructured YSZA cutting inserts and machining experimental procedures. After machining, the nanostructured cutting inserts are evaluated with scanning electron microscopy (SEM) and optical microscope, and compared with commercial ceramic cutting inserts under the same conditions.

Key words: Nanostructured materials; Cutting inserts; Machining.

1 Introduction

Due to their superior mechanical and chemical properties such as hardness, abrasive resistance, chemical inertness and high stability in high temperature, advanced engineering ceramics offer unique capabilities as tribological materials [1]. Advanced ceramics for tribological applications mainly include alumina (Al_2O_3), silicon nitride (Si_3N_4), silicon carbide (SiC), zirconia (ZrO_2) and SiAlON . One of the applications of these materials is cutting tools.

Cutting tools made of advanced engineering ceramics have been applied to high-speed finishing operations and machining of difficult-to-machine materials. However, due to their high brittleness and relatively low fracture toughness, the extent of the application of advanced ceramics in cutting tools has been limited. Materials with fine microstructures have been recognized to exhibit technologically attractive properties. When the grain size of a material decreases to a scale of tens of nanometers, one obtains a novel class of materials, called "nanostructured materials" (*n*-materials), which possess microstructures and properties different from their conventional counterparts [2]. It has been known that both hardness and toughness can be enhanced by the reduced grain size and unique microstructures in *n*-materials [3]. Therefore, the performance of the cutting tools made of *n*-materials is expected to improve.

This research investigates the cutting inserts made of nanostructured yttrium-stabilized

zirconia and alumina in turning of high speed M1 tool steel. After turning, the *n*-YSZA inserts are observed with a scanning electron microscope. Flank wear (VB) is measured with an optical microscope. For comparison, commercial alumina cutting inserts have been used in machining.

2 Experimental Procedures

Bulk *n*-YSZA was fabricated from its nanoscale powders. In the fabrication, a method called transformation-assisted consolidation (TAC) was used, which was similar to that for bulk nanoscale titania described by Liao *et al.* [4]. This method used high pressure and relatively low temperature in sintering to limit grain growth. The nanoscale grain size in powder is sustained under this condition and the density of the sintered materials is also enhanced.

The bulk *n*-YSZA was sliced and then ground with a diamond wheel on a high-precision surface grinder. The round corners of the cutting inserts were ground on a cylindrical grinder with a special grinding jig as shown in Fig. 1a. All the inserts were manually lapped and polished with diamond pastes to eliminate possible surface damage inherited from the previous grinding processes. The insert had dimensions of $12.7 \times 12.7 \times 4.8$ mm (or $\frac{1}{2} \times \frac{1}{2} \times \frac{3}{4}$ inches) and corner radius of 1.2 mm (or $\frac{1}{4}$ inches), conforming to the ISO specifications of SNG433. Fig. 1b shows the 3D view of the inserts and Fig. 1c is the cutting tool insert-holder assembly.

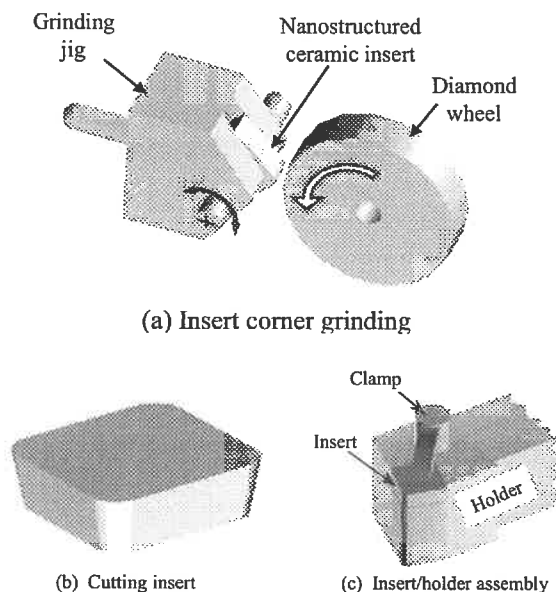
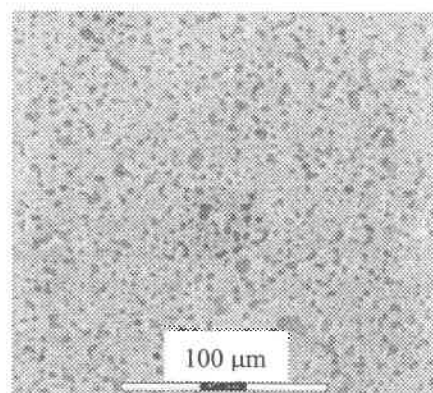


Fig. 1 Development of *n*-YSZA insert.



(a) *n*-YSZA insert

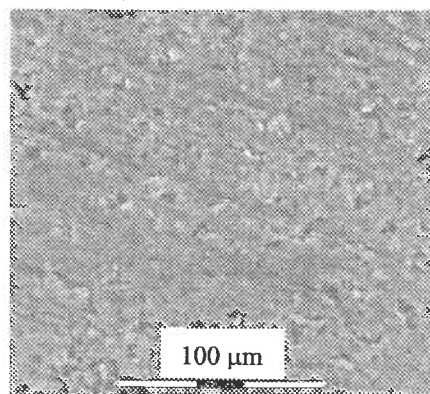


Fig. 2 Comparison between rake faces of the inserts before machining.

The rake face of the as-made inserts was observed with SEM. The micrograph in Fig. 2a clearly shows two phases: alumina uniformly embedded in yttrium-stabilized zirconia. Due to the lapping and polishing processes, grinding marks are invisible for the *n*-YSZA insert compared to the commercial alumina insert shown in Fig. 2b.

The *n*-YSZA inserts were used to machine high-speed M1 tool steel at a depth of cut of 0.76 mm, a feed of 0.14 mm per revolution and a cutting speed of 330 m/min. Machining was in dry condition without lubrication. The machining conditions are summarized in Table 1. The tool holder was MSRNR16-4D, which provided a 5° negative rake angle. The machining process was real-time monitored by a 3D force transducer. After machining, the inserts were observed with SEM to investigate wear mechanisms. Under the same machining conditions, the commercial alumina cutting inserts were used for comparison. Through the comparison, the differences in wear mechanisms between the nanostructured and commercial inserts were identified, which can provide valuable information for the manufacturing of nanostructured cutting inserts.

Table 1 Machining Conditions

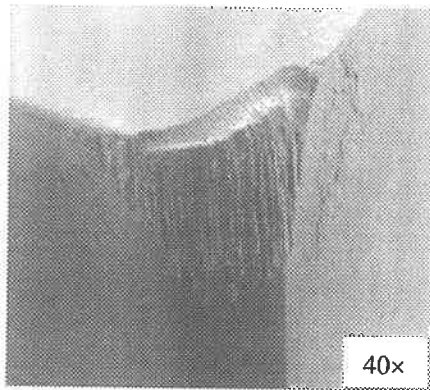
Workpiece Material	M1 high speed tool steel
Depth of cut	0.76 mm
Feed	0.14 mm/rev
Machining speed	330 m/min
Machining coolant	Dry

3 Results and Discussions

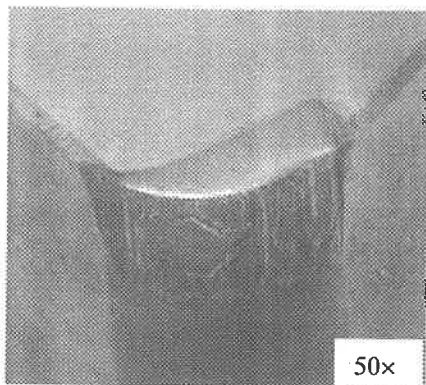
As stated above, this study was aimed at exploring the potential applications of nanostructured ceramics to cutting inserts. The performance of a cutting insert depends on its material properties and could be judged through its wear observations after machining. After 30 minutes of machining of high-speed M1 tool steel under the conditions in Table 1, both *n*-YSZA and commercial alumina inserts were observed with SEM for cutting edge wear and measured by the optical microscope for flank wear (VB).

Figures 3a and 3b show the overall pictures of the worn inserts. From the figures, it is observed that both inserts were severely worn down on their cutting edges. However, the *n*-YSZA insert had more severe wear than the commercial

alumina insert: deeper wear crater and wider flank wear land. Also, the worn rake face of the *n*-YSZA insert is rougher than that of commercial insert. Obvious chipping is observed on the cutting edge of *n*-YSZA insert.



(a) Worn *n*-YSZA insert



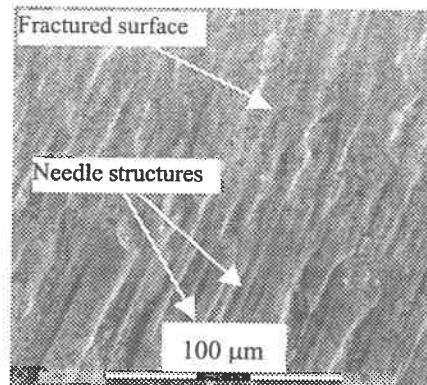
(b) Worn commercial alumina insert

Fig. 3 SEM observations of cutting edges subjected to machining.

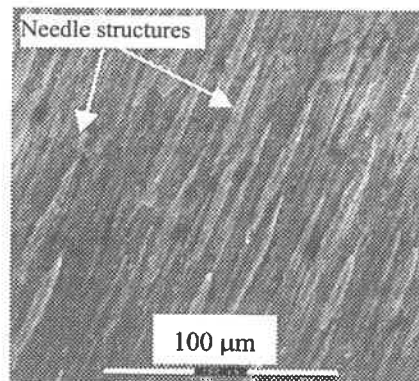
A closer look at the worn rake faces of two inserts (Fig. 4) shows that both fracture and ductile flow are observed on the *n*-YSZA insert while only ductile flow is observed on the commercial alumina insert. Additionally, for the *n*-YSZA insert, fracture seems to be the dominant wear mechanism. Ductile flow formed needle structures, which increase the wear resistance, on the rake surfaces of both inserts. However, more and smaller needle structures are observed on the commercial insert than on the *n*-YSZA insert.

Ploughing combined with fracturing resulted in rough flank face with deep wear grooves on the *n*-YSZA insert, as shown in Fig. 5a. Attrition contributed to gradual wear, flaking, of the

commercial insert, as shown in Fig. 5b. Flaking caused layer-by-layer material wear, and therefore smaller and slower flank wear for the commercial cutter. The average flank wear (VB) are compared in Fig. 6 for the inserts, from which it is found that under the same machining conditions, the *n*-YSZA insert had more severe flank wear.



(a) *n*-YSZA cutter



(b) Commercial alumina insert

Fig. 4 Wear characterization of rake faces of the worn inserts.

4 Conclusions

The cutting inserts of *n*-YSZA have been evaluated in machining high speed M1 tool steel, and are found to be inferior to the commercial alumina inserts. Nevertheless, this research is the first attempt to apply *n*-materials to cutting inserts. Further investigations are needed.

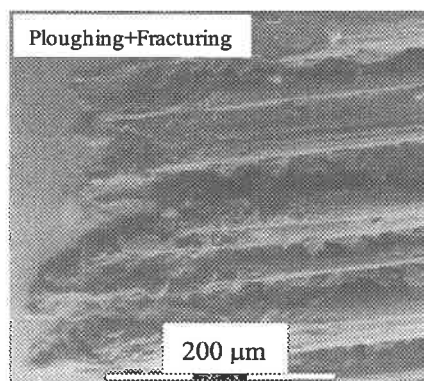
Acknowledgments

The authors thank NanoPac Technologies, Inc. for nanostructured yttrium-stabilized zirconia and alumina material. Thanks are also due to

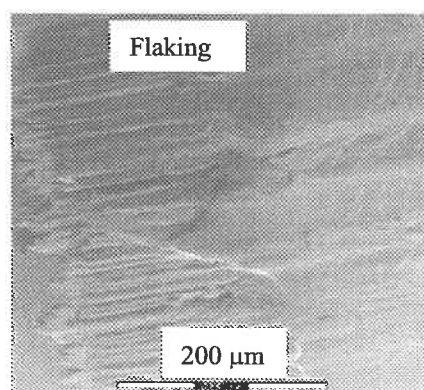
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(a) *n*-YSZA insert



(b) Commercial alumina insert

Fig. 5 Flank wear observation.

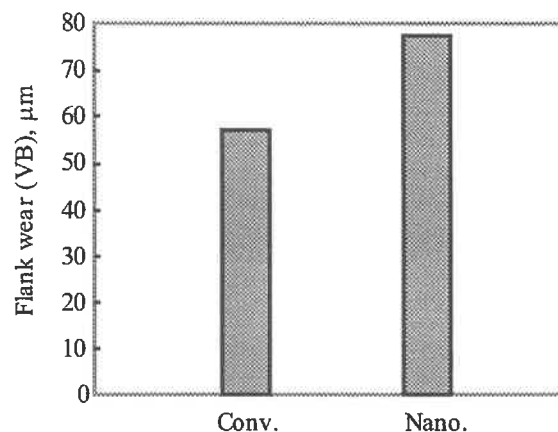


Fig. 6 Average flank wear (VB) comparison

Nonlinear Effects in Precision Machining Engineering Materials

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Abstract: The dynamic surface generation in precision turning processes is modelled and simulated by an integrated modelling approach. The nonlinear models, such as regenerative chatter model, tool wear model and shear-localized chip model, combine with the dynamic cutting force model, the machine tool response model and the surface topography model to form the whole modelling approach. Matlab Simulink is used to interactively perform the simulation in a user-friendly, effective and efficient manner. The models and simulation results are evaluated and validated by precision turning trials. The machining process nonlinearities degrade the machined surface finish. The formation of shear-localized chips has very significant effect on the generation of machined surface.

Key words: precision turning, nonlinearity, chatter, surface finish

1 Introduction

The demands for high precision products have been continuing to increase over the last few decades due to the increased global competition. Achieving better understanding of the precision machining processes is essential to further improve the machining accuracy along with developing high precision machine tools. The precision turning processes are highly dynamic because of the high cutting speed. Furthermore, strong nonlinearities can arise from the cutting processes and the behaviour of the machine-tool, and these nonlinearities have unpredictable effects on the process dynamics and the surface accuracy. If these effects could be accurately modelled and thus controlled, significant improvement in performance might be attained by including control and compensation algorithms in machine tool controllers.

An early nonlinear model that accounts for the separation of the tool and workpiece is that given by Doi and Kato [1]. Hanna and Tobias introduced a nonlinear differential difference equation to describe regenerative chatter in milling processes, in which the nonlinearities of the machine tool structure are included [2]. In recent years, some new methods have emerged in the study of nonlinear dynamics of machining processes. Stépán used a four-dimensional phase-space representation embedded in the infinite-dimensional phase space of regenerative chatter to investigate the chatter vibrations of an industrial machine tool [3]. Nonlinear dynamics techniques such as constructing bifurcation diagrams and Poincaré map are employed by Wiercigroch and Krivtsov to ascertain dynamics response for both the deterministic and stochastic models of friction chatters [4]. Dynamics and stability of milling operation with cylindrical end mills were discussed by Balachandran, who proposed a unified mechanics-based model allowing for both regenerative and loss-of-contact effects [5]. Davis and Burns proposed a novel nonlinear dynamics approach to predict the transition from continuous to shear-localised chip formation [6]. A sound foundation has been laid out by those investigations.

In this paper the dynamic surface generation in precision turning processes is investigated with an integrated modelling approach. The effects of nonlinearities are studied by the simulation and machining trials.

2 Modelling precision turning process with nonlinearities

In the precision turning process the workpiece and cutting tool can be simplified as a second-order spring-damper vibratory system in the X, Y and Z directions. For the sake of simplicity, the machine tool structural nonlinearities are ignored. The workpiece is assumed to be rigid. The whole cutting system can be described as:

$$\begin{bmatrix} x(t) \\ y(t) \\ z(t) \end{bmatrix} = \begin{bmatrix} G_{xx} & G_{xy} & G_{xz} \\ G_{yx} & G_{yy} & G_{yz} \\ G_{zx} & G_{zy} & G_{zz} \end{bmatrix} \begin{bmatrix} F_x(t) \\ F_y(t) \\ F_z(t) \end{bmatrix} \quad (1)$$

where $x(t)$, $y(t)$ and $z(t)$ are the dynamic displacements of the cutting tool in the X, Y and Z directions. G_{ab} is the corresponding response of the tooling structure in a -th direction due to the force acting in the b -th direction when the other two force components are zero (a and b stand for the X/Y/Z direction respectively). $F_x(t)$, $F_y(t)$, $F_z(t)$ are the dynamic cutting forces in the X, Y and Z directions. t is time.

The whole modelling approach can be illustrated in Fig. 1. The machining variables, tooling characteristics and nonlinearities in the machining process are inputs to the dynamic cutting force model. The three dimensional cutting forces are generated in the light of the interactions between the cutting tool and workpiece in the cutting zone, which will act on the machine tool and tool structure and make the real tool path away from the ideal tool path. The real

tool path can be calculated by the machine tool response model. Based on the real tool path, the machined surface can be generated by calculating the intersection points of the tool path sequence in surface topography model.

The machine tool response model is built based on the modal analysis of the machining structure. The surface topography model can be acquired by calculation of the intersection points of the tool paths. The details can be seen from another paper of the authors [7]. This paper only focuses on the nonlinear models and dynamics cutting force model.

Regenerative chatter is one of primary sources of vibrational instabilities in precision turning. It is excited by the variation of cutting forces due to cutting a wavy surface left by the previous cut. The regenerative chatter can be modelled through the variation of the real chip thickness and chip width. Considering the regenerative chatter, the dynamic chip thickness and dynamic chip width can be expressed as:

$$c_h(t) = f - [z(t) - z(t-T)] \quad (2)$$

$$c_w(t) = d_c - r + x(t-T) - x(t) \quad (3)$$

where f is the feed rate, r is the tool nose radius and d_c is the depth of cut. $x(t-T)$ and $z(t-T)$ are the tool positions at previous revolution of cut (time $(t-T)$).

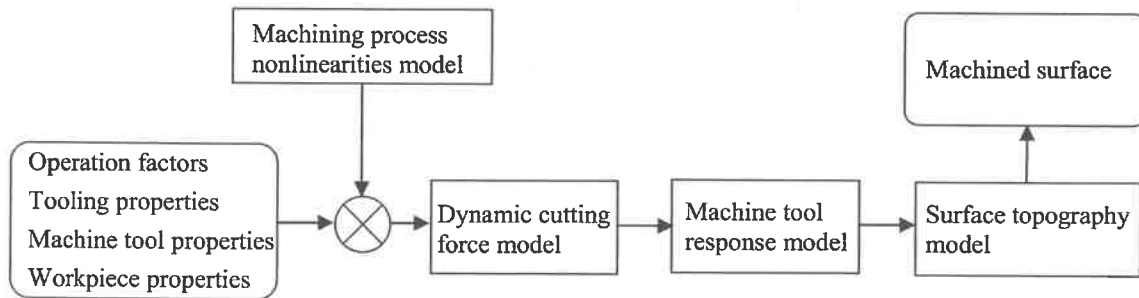


Fig. 1 Illustration of modelling approach for surface generation.

Chip formation is a highly dynamic and nonlinear process involving a very complex thermoplastic flow of workpiece material. Experiments show that, for most ductile metals, as the cutting speed increases monotonically in orthogonal cutting, a transition takes place from continuous to shear-localized chip formation in the flow field of the material being cut. The research carried out in NIST indicates that the formation of shear-localized chips is periodic at high cutting speed [6]. The shear stress of the material under the tool tip will vary from 0 to a maximum value because of the period thermal softening and straining harden of the workpiece with the formation of a series of segmented chips. The real shear stress in cutting process can be expressed as:

$$k_s = \text{mod}\left(\frac{V_c t}{\Delta S}\right) k_y \quad (4)$$

where k_y is the yield stress of the workpiece material. ΔS is the average distance between the shear-localized chips.

Tool wear is another nonlinear source in the turning process. The flank tool wear will result in the rubbing of the tool nose on the workpiece surface and bring forth the so called stick-slip oscillations. Tool wear land brings about a zero or sometimes negative clearance angle depending on the relative tool motion with respect to the workpiece. So the real clearance angle of the cutting tool in the cutting process can be modelled as:

$$\beta(t) = \beta_0 c(t) \quad (5)$$

where $c(t)$ is an acceleration function which describes the variation of clearance angle due to tool wear. β_0 is the initial clearance angle of the cutting tool.

The usage of coolant will change the friction angle or friction coefficient between the cutting tool and workpiece. So the real friction coefficient can be modelled as:

$$u(t) = u_0 r(t) \quad (6)$$

where $r(t)$ is a pulse function. μ_0 is the initial friction coefficient between the cutting tool and workpiece.

In the cutting zone, the cutting forces will act on the tool rake face, cutting edge and flank face. Accumulating the forces acted on the three zones, there will be the dynamic cutting forces in the three directions expressed as:

$$\begin{cases}
F_x(t) = K_{tc}[c_w(t)c_k(t) + d_c\sqrt{r^2 - (r - d_c)^2}] + K_{te} \cdot [c_w(t) + r\theta_r] + K_{f1}[\sin(\frac{\beta(t)}{2}) \\
\quad - \mu(t)\cos(\frac{\beta(t)}{2})] + K_{f2}[\sin\beta(t) - \mu(t)\cos\beta(t)] \\
F_y(t) = [K_{rc}c_k(t) + K_{re}] \cdot c_w(t) + K_{fe}\sin\theta_r r + (K_{fc} - K_{rc})\cos\theta_r d_c\sqrt{r^2 - (r - d_c)^2} \\
F_z(t) = [K_{fc} \cdot c_k(t) + K_{fe}] \cdot c_w(t) + (K_{fc}\cos\theta_r + K_{rc}\sin\theta_r)d_c\sqrt{r^2 - (r - d_c)^2} + K_{fe}r\sin\theta_r \\
\quad + K_{f1}[\cos(\frac{\beta(t)}{2}) + \mu\sin(\frac{\beta(t)}{2})] + K_{f2}[\cos\beta(t) + \mu\sin\beta(t)]
\end{cases} \quad (7)$$

where θ_r is the intersection angle for two continuous tool paths.

3 Simulations and experimental trials

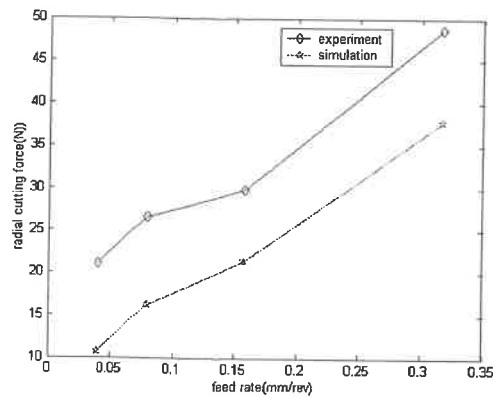
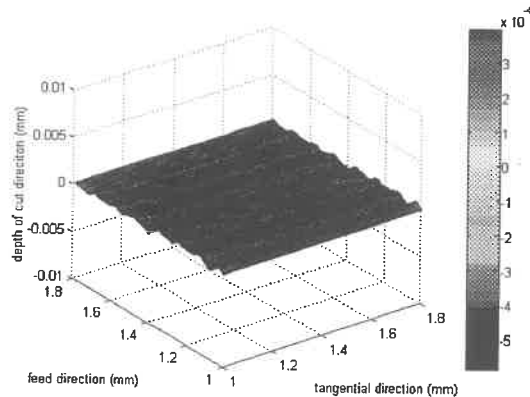


Fig 2. Simulated and measured radial cutting forces.

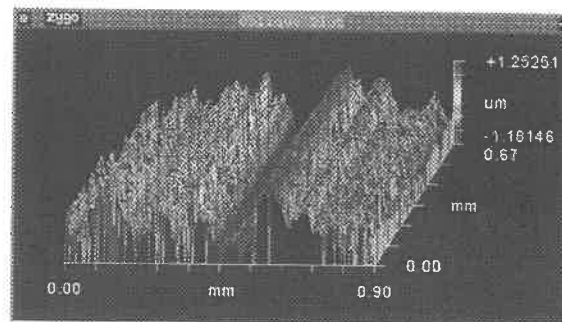
forces in the radial direction when the cutting is undertaken at the conditions: spindle speed = 1400 rpm, feed rate = 0.0397~0.3175 mm/rev and depth of cut = 0.1mm. It can be seen that the simulated cutting forces are well agreed

The whole machining dynamics model is implemented in a Matlab simulink environment, in which the nonlinearities are easily to turn on and turn off. Precision turning trials are carried out on a CNC lathe to validate the model and simulation. The dynamic cutting forces are measured by a Kistler dynamometer, 9257BA, on which the cutting tool is mounted. The machined surfaces are measured by the Zygo Newview 5000 optical microscope. The aluminium alloy bar with 50 mm diameter is turned in the experiments using a carbide insert with nose radius = 1.2 mm, initial side rake angle = 5°, back rake angle = 10° and clearance angle = 10°. The spindle speed varies from 770 rpm to 1000 rpm, feed rate varies from 0.0397 mm/rev to 0.3175 mm/rev and depth of cut is set at 0.1mm.

Fig. 2 shows the simulated and measured cutting forces in the radial direction when the cutting is undertaken at the conditions: spindle speed = 1400 rpm, feed rate = 0.0397~0.3175 mm/rev and depth of cut = 0.1mm. It can be seen that the simulated cutting forces are well agreed



(a) Simulated surface



(b) Machined surface

Fig. 3 Comparison between the simulated surface and the machined surface.

($f = 0.1588$ mm/rev, $n = 1400$ rpm, $d_c = 0.1$ mm, $r = 1.2$ mm)

with the measured results (about 34.8% lower than the measured results). It also shows the tendency that the radial cutting force increases with the increment of the feed rate. When the feed rate 0.1588 mm/rev is applied and the nonlinear models are turned on, the simulated machined surface and the measured surfaces are shown in Fig 3. The

direction of lay is evident in both the simulated surface and the experimentally generated surface. The root-mean square deviation S_q of the simulated surface is $0.24\text{ }\mu\text{m}$, which is close to the experimental result $0.96\text{ }\mu\text{m}$. The difference between the simulation result and experimental result may be caused by the estimated static machining structural parameters. But the simulation results are still in the reasonable deviation scale.

4 Discussions

In order to study the effects of the machining process nonlinearities, simulation is run with the same operation condition as the simulation run in Fig. 3 but with the nonlinear models turned off. Fig. 4 shows the simulation result. Comparing with the result in Fig. 3, it can be seen that the machining process nonlinearities degrade the surface finish as in Fig. 3 the R_q value nearly doubles the R_q value in Fig. 4.

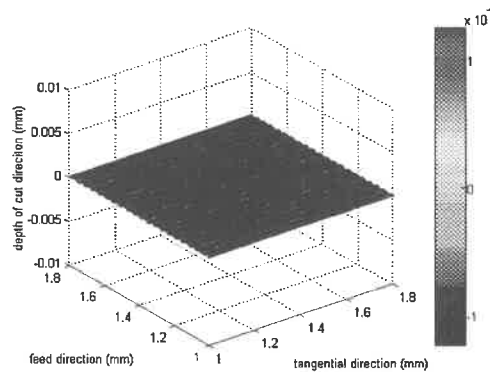


Fig. 4 Simulated surface without nonlinearities.

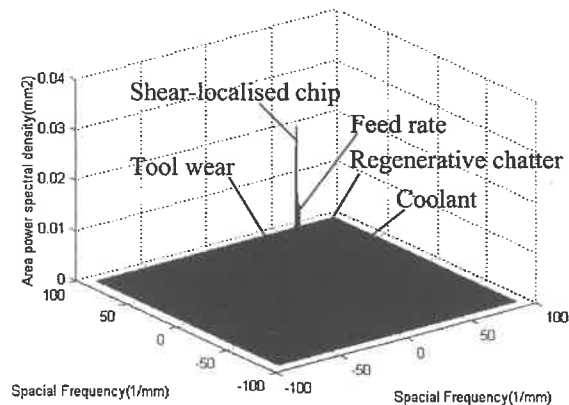


Fig. 5 APSD of the machined surfaces.

The significance of the machining process nonlinearities is also studied by the spectral analysis technique. Fig. 5 shows the area power spectral density of a machined surface. It can be found that the shear-localised chip is the most dominant factor for the machined surfaces, which special frequency in feed direction is 10.3 cycle/mm . Feed rate is also significant Its frequency is $1/f = 1/0.1588 = 6.29\text{ cycle/mm}$. The regenerative vibration, tool wear and usage of coolant are less significant, but still leave their marks on the machined surface.

5 Concluding remarks

The generation of machined surface in the precision turning process is modelled by an integrated approach, in which the machining process nonlinearities have been modelled in the light of their contributions to the dynamic cutting force. The simulation is performed in Matlab simulink. The experimental trials results agree very well with the simulation results. The machining process nonlinearities degrade the machined surface finish. The formation of shear-localised chips is the most significant nonlinear factor affecting the generation of machined surface.

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ELLIPTICAL VIBRATION ASSISTED DIAMOND TURNING

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INTRODUCTION

Fabrication of optical quality surfaces on non-ferrous materials can be produced with single crystal diamond tools and precision turning machines. To add ferrous and brittle materials to the list of diamond turnable materials, a process known as Elliptical Vibration Assisted Machining (EVAM) is being developed. The PEC has built an EVAM system known as the UltraMill [1] that combines a small elliptical tool motion ($40\text{ }\mu\text{m}$ horizontal by $6\text{ }\mu\text{m}$ vertical at 600 v excitation) with the linear motion of standard diamond tool (1 mm nose radius). EVAM reduces cutting forces and tool wear by reducing chip thickness and keeping the tool tip out of contact with the workpiece for 75% of the time. These characteristics increase tool life and expand work piece material compatibility with diamond tools.

SURFACE ROUGHNESS MEASUREMENTS

The surface roughness of the EVAM machining experiments was compared to a theoretical prediction for grinding surface finish developed at the PEC by Storz [2]. This comparison is illustrated in Figure 1. Here the areal RMS roughness as measured on a Zygo New View is plotted as a function of the Horizontal Speed Ratio (HSR) for a 1 KHz EVAM frequency. The HSR is the ratio of the maximum speed of the elliptical motion to the surface speed of the part and is proportional to the distance that the part moves for each elliptical cycle of the tool. The general trend of the measured roughness curve follows the theoretical behavior. The minimum measured surface roughness is 6.5 nm RMS. While Figure 1 was generated at 1 KHz, the speed of the EVAM motion (1-4 KHz) does not seem to affect the minimum roughness; that is, the vibrations associated with the tool motion do not adversely affect the surface roughness.

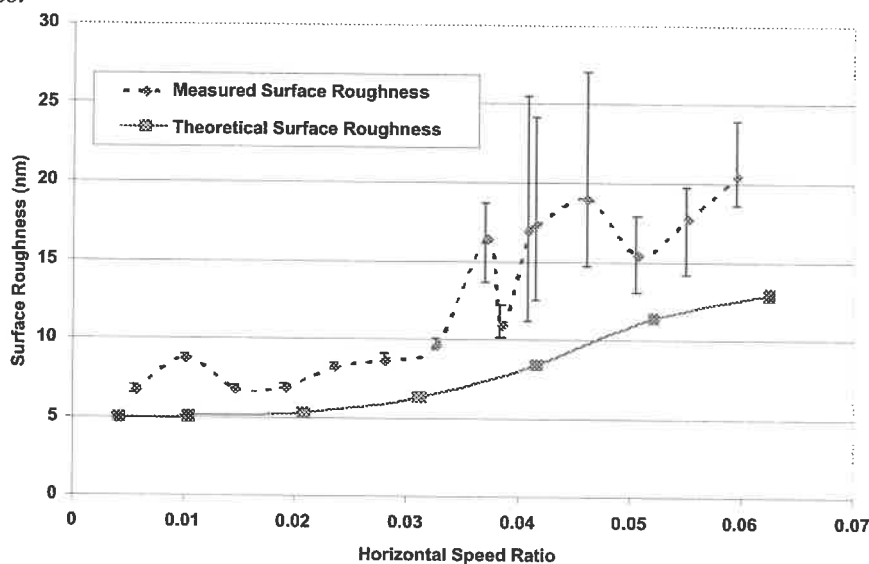


Figure 1. Measured and Theoretical Surface Roughness on PMMA
(1 KHz, 21.4 rpm, 0.25 mm/min, 12 $\mu\text{m}/\text{rev}$)

SiC MACHINING EXPERIMENT

The groove cutting experiment done previously at the PEC proved that the EVAM process could be used to machine SiC; however, creating an optical surface is a more challenging problem. To demonstrate the possibilities, the UltraMill was used to machine a surface with a 1 mm outer radius and a 0.5 mm inner radius. The fracture data from the groove cut experiment showed that damage free surfaces could be

machined if the chip thickness is less than 20 nm. The UltraMill was operated at the same conditions as in Figure 1 with the cross feed of the tool set to 12 μm per revolution. The SiC specimen was vacuum mounted to the DTM spindle and the part rotated at 1 RPM to create a maximum HSR of 0.08% at a radius of 1 mm. Once supported on the vacuum chuck, the surface flatness was less than 600 nm over the area to be machined.

Machining Procedure The UltraMill was operated at 1 KHz for 40 minutes to allow the temperature to equilibrate. After the equilibration period, the tool was brought into contact (touch off) with the rotating specimen at a radius of 0.4 mm. This was done by positioning the tool tip to within several micrometers of the surface and moving in 100 nm increments until a chip was created as observed with a long-range microscope. The oil jet was activated and the part program was run. The machining process required 44 minutes, during which the temperature of the UltraMill piezo stacks changed by -0.4°C . The goal to avoid depths of cut greater than 1.2 μm was achieved (equivalent to a chip thickness less than 20 nm) and the measured depth ranged from 30 nm to 418 nm with an average depth of 300 nm.

Surface Measurements Figure 2 is an SEM micrograph of the machined surface of the SiC. The nominal cutting distance was 196.25 mm and the volume removed was $7 \times 10^{-4} \text{ mm}^3$ with a removal rate of $15,900 \mu\text{m}^3/\text{min}$. The average of the tangential and radial profiles and surface measurements are plotted as a function of radial position in Figure 3. The machined SiC surface has an average roughness of approximately 8 nm RMS. The theoretical RMS roughness of the machined surface is dominated by the cross feed and has a value of 5.5 nm. The theoretical roughness along the groove bottom is 1 angstrom. This value is much less than the spindle error motion and therefore is not visible. The value shown in Figure 3 is equal to the average spindle error.

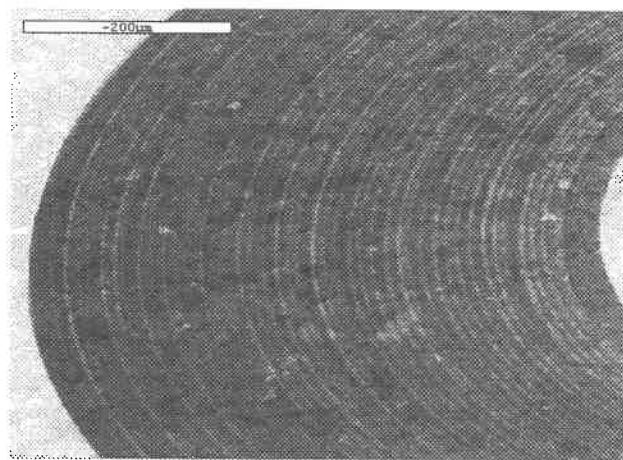


Figure 2. SEM Micrograph of SiC Surface

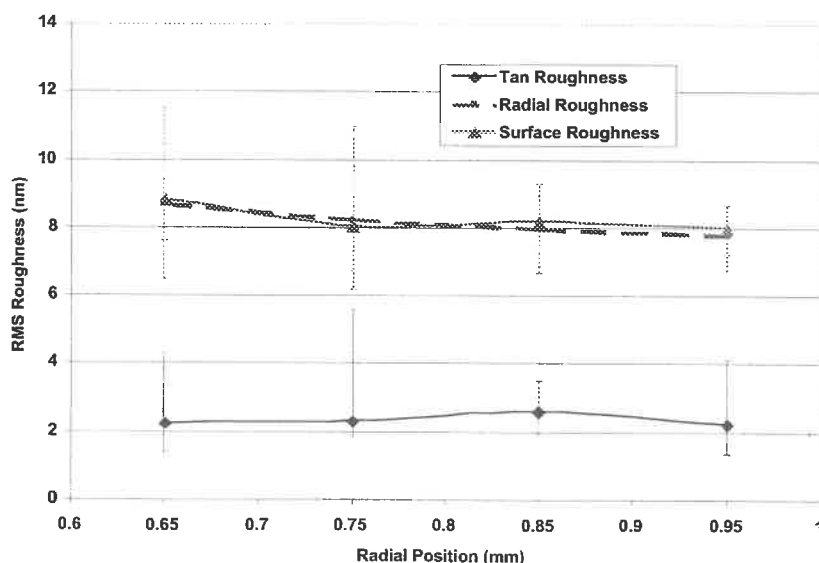


Figure 3. Surface Measurements of Machined SiC CVD
(1 rpm spindle speed, cross feed = 12 μm , upfeed = 40 nm, frequency = 1 KHz)

The radial and surface measurements in Figure 3 are similar in magnitude, 8 nm RMS, suggesting that the overall surface roughness is mostly a result of cross feed features. This was expected based on the selected machining conditions and the large difference between the cross-feed and up-feed rates.

Figure 5 is a Zygo contour image of the machined surface at a radius of 0.85 mm. The cross feed marks can be clearly seen at 12 μm spacing. Normally these would be connected by vertical lines due to the upfeed motion created by the elliptical motion of the tool. However, for this experiment, the upfeed rate of 40 nm/cycle is beyond the resolution of the instrument. What are visible are lower frequency vibrations attributed to spindle and DTM slide motion. A trace along one of the grooves has amplitude of 2.5-5 nm at a frequency of 25 Hz. Surface roughness along the machined groove is 1.9 nm RMS. Some surface damage was noted and can be seen along the bottom of one of the up feed grooves.

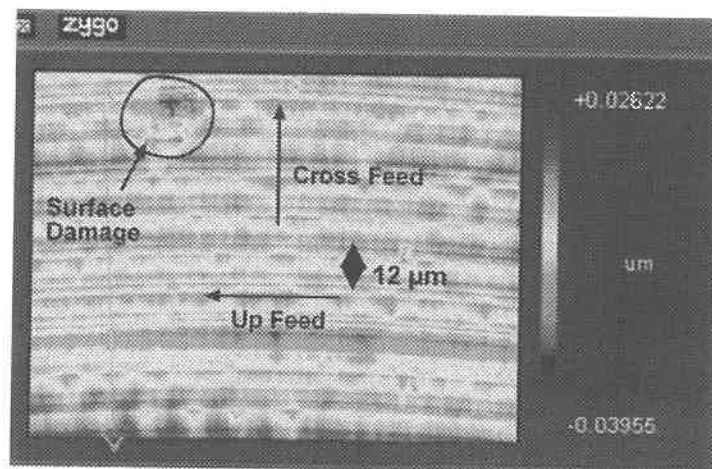


Figure 4. Contour Plot of EVAM Surface in SiC (HSR=0.08%)

A profile trace in the cross feed direction (Figure 5) provides another view of the surface. This scan reveals a cross feed period of approximately 11.9 μm and a local feature height of 16 μm . These values are similar to the theoretical 12 μm cross feed and 18 μm peak-to-valley feature height. The cross feed features have distinct, repeated characteristics which are believed to be the result of interaction between the tool edge and the SiC surface during machining. These features appear to be small fractures or damage sites on the tool edge, less than 5 nm deep.

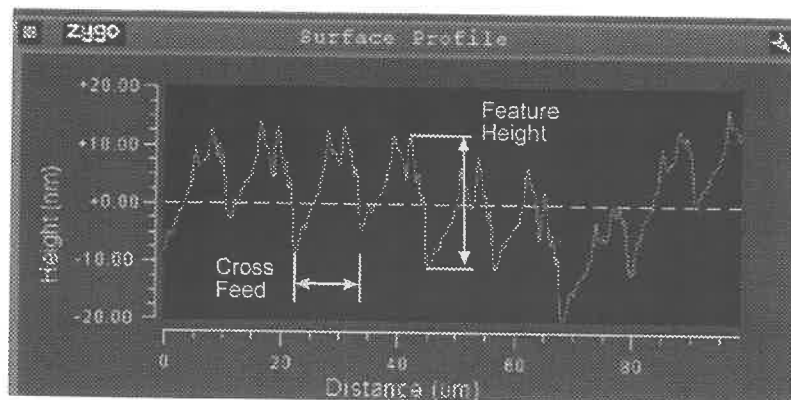


Figure 5. Radial Profile of SiC Surface (Cross Feed = 12 $\mu\text{m}/\text{rev}$)

TOOL WEAR

SEM micrographs of the tool tip before and after the machining process reveal mild wear. The cutting edge was imaged with the SEM before machining the SiC and is shown on the left side of Figure 6.

There appears to be no visible damage to the edge. After machining the SiC, the tool edge was again inspected with the SEM. No indication of tool wear could be found on the rake face; however, the flank face revealed features shown at the right of Figure 6. The width of the worn area is the same as the theoretical contact length of $25\text{ }\mu\text{m}$ for a 1 mm radius tool at a depth of 300 nm . This correlation suggests that this wear area was created during the machining process.

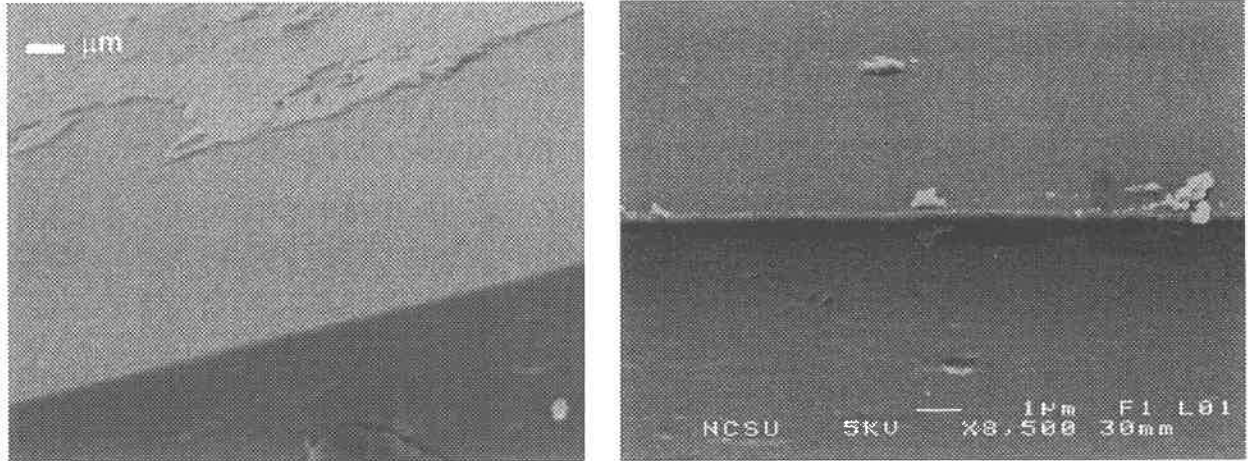


Figure 6. Unworn (left) and Worn (right) regions on the tool edge

A close up view of the flank wear region in Figure 7 shows wear patterns and striations in the upfeed direction as expected. The length of the wear land is approximately 400 nm . Given this wear land and a clearance angle of 10.4° , a rake face recession of 74 nm is estimated. This calculation assumes a flat wear land that may not be the case for the EVAM process.

X.3 CONCLUSIONS

The UltraMill is a rugged, reliable design for creating surfaces using elliptical vibration assisted machining. It has two major advantages over previous designs; first, the elliptical motion has a significant vertical component that can be used to control the thickness of the chip; and second, motion of the tool is below the first natural frequency so the speed is easily adjustable from 1-5 KHz. This device has been used to machine a variety of materials, from aluminum to plastics to SiC. It can produce optical quality surfaces and the accompanying tool wear for difficult to machine materials – steel and ceramics – is less than conventional diamond turning.

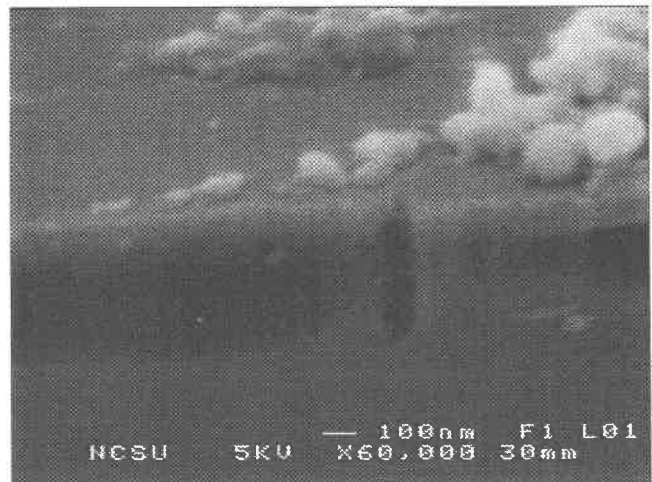


Figure 7. Worn Edge on Diamond Tool

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Analysis of accuracy and cutting forces in groove cutting

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1.Introduction

By the improvement of machine tools and cutting tool materials , cutting efficiency has been increased in late years. Especially , carbide materials and coatings enabled the cutting of hardened steel. Ball end mills are commonly used for machining of dies. In order to avoid the interference of tool and work , long tools are used. During the machining of dies , cutting forces acting on an end mill affect the cutting accuracy due to the bending of the tool as shown in Fig.1. In order to predict the accuracy of machined surface , it is necessary to predict cutting forces.

There have been many researches on cutting forces while slotting by ball end mills with large depth of cut¹⁾²⁾. But it is necessary to select small depth of cut in the machining of hardened materials. While slotting by a ball end mill in the condition of small depth of cut , geometrical shape of uncut chip becomes complex as shown in fig.1.

In the previous paper , a new method to predict the cutting forces during slotting by round trip cutting. At first , uncut chip thickness is calculated under the assumption that the end mill is fully immersed. Then the uncut chip thickness of the part which is already cut and stock material does not exit is set to be 0. This paper reports the results of an application of this method to analyze the cutting forces during contour milling or helical milling. During contour milling and helical milling , there exist not only axial depth of cut but also radial depth of cut. Thus the shape of chip becomes complex.

2.Experimental Procedure and Conditions

A vertical machining center (YASDA YBM-640V) was used for the cutting experiments. Maximum spindle revolution is 20000rpm , and a HSK shank which has high repetition accuracy in tool exchanging was adopted. Maximum table speed is 10000mm/min for each x , y , z axis. The run out of end mill was arranged to be within 5 μ m.

Cutting conditions are shown in Table 1. Experimental setup is shown in Fig.2. Piezo-electric dynamometer (Kistler 9257B) was used for measuring cutting forces ,

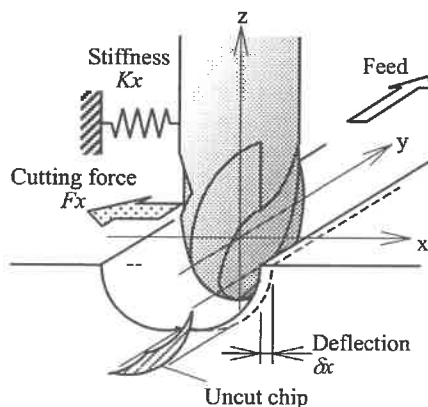


Fig.1 Deflection of end mill and cutting force

and specimen was fixed on the dynamo meter.

Table 1 Cutting conditions

Tool	Ball End Mill $\phi 10$ R5 WC(TiAlN coated)
Work	HPM1 (HRC40)
Cutting Conditions	Spindle rotation 3000min^{-1} Axial depth of cut (Ad) 0.1mm Feed rate (f_z) 0.1mm/tooth
Coolant	Oil

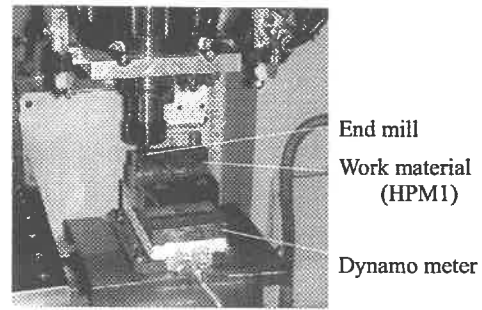


Fig.2 Experimental setup

3. Analytical and Experimental results of cutting forces

3.1 Cutting forces of contour milling

In the previous paper, a new method was proposed to predict the cutting forces of slotting by calculating the chip flow direction by Stabler's law and measuring contour shape of cutting edges of ball end mills. And the chip load of cutting edge that exists in the area of previous path is set to be 0 as shown in Fig.3. By using this method, it became possible to predict the cutting forces of a ball end mill without complex geometrical calculation of boundary conditions.

The method shown in previous paper is the case of slotting that ball end mill is driven in axial and feed direction. And the radial depth of cut is equal to the diameter of the tool. In this paper, a method to calculate when the radial depth of cut becomes smaller than tool diameter is proposed. During the contour milling, ball end mills are fed in radial direction as shown in Fig.4. The shape of uncut chip is shown in Fig.5. The dotted lines shown in Fig.5 are the contour of the previous cutter location. In this method, cutting edges are divided into small elements. And chip load is once calculated. The chip load of the element that exists in the area which is cut by the previous path should be 0.

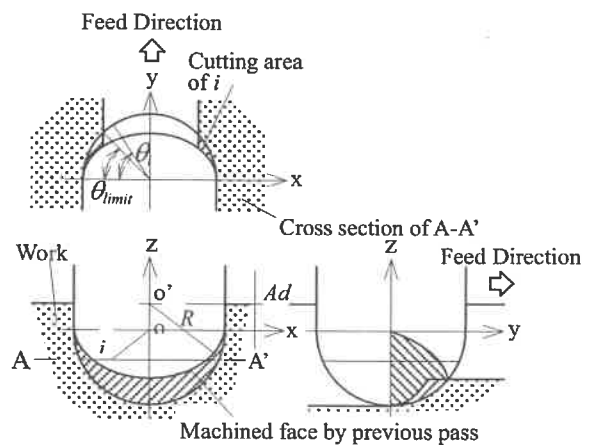


Fig.3 Schematic drawing of cutting area

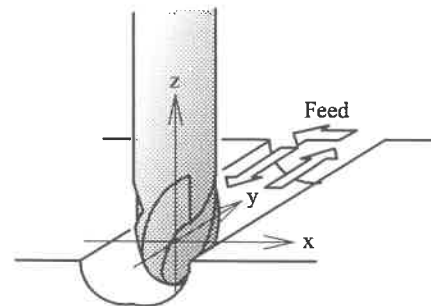


Fig.4 Deflection of end mill and cutting force

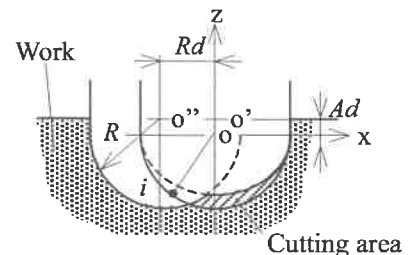


Fig.5 Schematic drawing of rigid cutting area

Thus, $(Rd^2 - x_i(\theta)^2) + z_i(\theta)^2 < R^2 \Rightarrow \Delta a_i = 0$

x_i and z_i are the coordinates of the i th element. o'' is the center of ball end mill before being fed in radial direction. o' is the center before being fed in axial direction. Fig.6 shows the flow chart of the analyzing method.

3.2 Evaluation of the cutting forces

Cutting forces predicted by this method and measured are shown in Fig.7 and Fig.8. Fig.7 shows the cutting forces during down milling and Fig.8 shows the forces during up milling. Cutter diameter is 10mm, radial depth of cut is 5mm, axial depth of cut is 0.1mm. These figures show that the wave form of the predicted cutting forces represents the characters of measured cutting forces. Although the amplitude of down milling cutting force is small, amplitude of up milling force is larger at the rotation angle $\theta = \pi/2$.

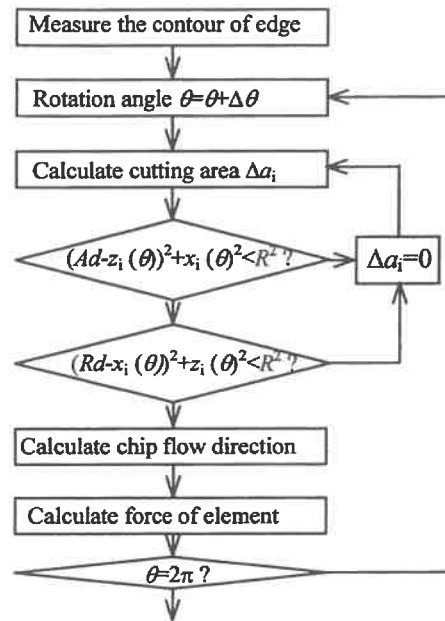


Fig.6 Procedure of the analysis

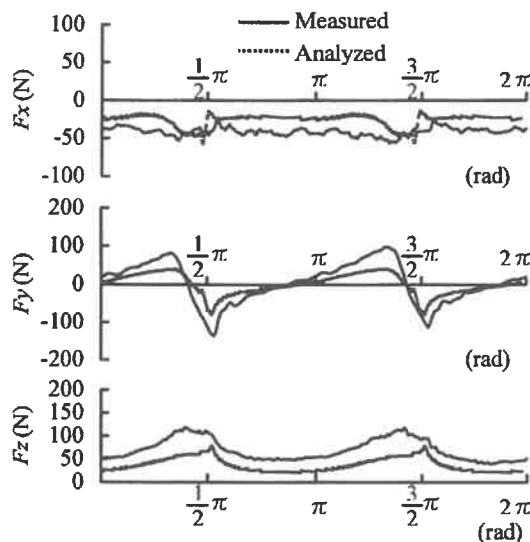


Fig.7 Cutting forces in the case of contour line path (Down milling)

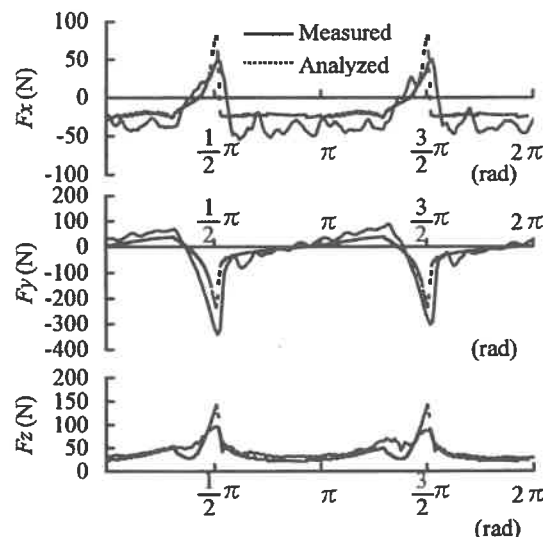


Fig.8 Cutting forces in the case of contour line path (Up milling)

3.3 Relationship between cutting forces and accuracy

In the cutting of groove by round trip milling or contour milling, the cutting force of radial direction determines the width of the groove. In this method, the rotation angle is defined to be $\pi/2$ when the outer cutting edge cuts the work as shown in Fig.9. Thus the bending caused by the cutting force at $\pi/2$

determines the cutting accuracy. In the up milling , cutting force F_x acts in the $x+$ direction. F_x acts in the $x-$ direction in down milling. Width error of the groove e is given as follows;

$$e = 2\delta_x = \frac{2F_x(\theta = \pi/2)}{K_x}$$

K_x is the stiffness of the tool in radial direction.

Fig.10 shows the width error of the groove in down milling. Measured cutter diameter is 0.980mm and the runout is arranged to be within 0.005mm. Tool length is 2.5mm and measured stiffness of the cutter K_x was 0.663×10^6 N/m. This figure shows that width of the groove becomes smaller in down milling than designed groove width.

4. Conclusions

In this paper , a method to analyze the cutting forces of ball end milling is proposed and the predicted cutting forces are evaluated. Results are clarified that:

- (1) By the procedure of setting the chip load 0 , it is possible to calculate complex cutting area.
- (2) It is possible to predict the width error in groove cutting by contour milling.

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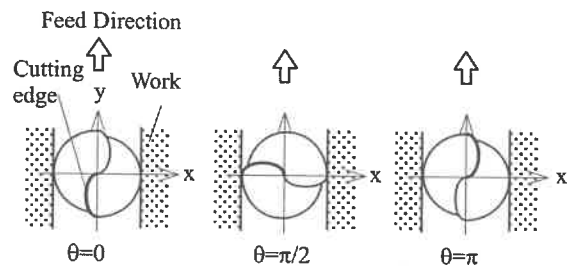


Fig.9 Relationship between rotation angle and position of cutting edge

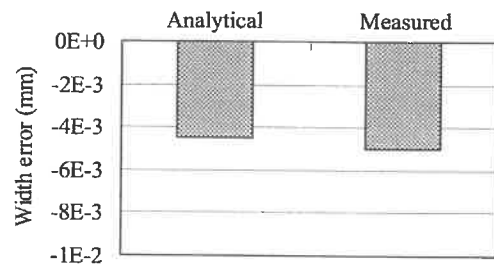


Fig.10 Error of analytical and measured slot width

Development of a Tunable Ultrasonic Vibration-Assisted Diamond Turning Instrument

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Introduction

Precision machining of brittle materials has until recently been primarily conducted by grinding and polishing due to excessive tool wear and surface fracture in traditional processes. In 1991, Moriwaki and Shamoto¹ introduced vibration-assisted machining of stainless steel, a process whereby a single point tool is vibrated at frequencies as high as forty kilohertz. This process reportedly reduces friction and therefore cutting force, extending tool life and significantly improving surface finish.² The variations of this technique allow for the vibration directions to include the cutting direction, the thrust direction, and the feed direction.¹ For ultrasonic vibration-assisted diamond turning, typical frequencies of tool tip vibrations are on the order of forty kilohertz and amplitudes are up to fifteen micrometers. These severe requirements have limited the previous designs to mechanical resonator systems with fixed operating frequencies and amplitudes due to excessive heat generation in piezoelectric actuators and limitations of system mechanical dynamics.

Drive Method

This paper describes a device under development at the University of North Carolina at Charlotte which provides for direct drive, ultrasonic linear vibration of the tool in which stroke and frequency can be easily varied at frequencies up to forty kilohertz and amplitudes up to seven micrometers. The novel characteristic of this device is the adjustable nature that will allow for correlations between the cutting parameters such as tool wear and surface finish. In order to achieve this tunable capability the resonator has been abandoned for a system with the first mechanical resonance well beyond forty kilohertz. The design incorporates multiple piezoelectric stack actuators and an offset impulse drive method*. Thermal issues associated with high-frequency piezoelectric motions typically limit the attainable frequency and displacement, but this issue is addressed in this design by coupling numerous actuators to the drive components. The drive system consists of multiple actuators in a series or parallel arrangement with each actuator operating at a fraction of the output frequency. An actuator has been developed using three "ganged" piezoelectric stacks, but the technique can be applied to any number of actuators by dividing the excitation signal into equal components. As an example, in the latest device design the drive system consists of three piezoelectric stack actuators driven by three signals separated by 120 degrees. For a desired output of 39 kHz the input signal to each actuator is 13 kHz. At this time the frequency requirements have been achieved and the displacement is up to seven micrometers at forty kilohertz.

The signal for each actuator is chosen to be a square-wave pulse-train supplied from a custom pulse generator with a variable pulse frequency and variable pulse width denoted in Figure 1 as $1/T$ and b respectively. A fixed phase shift is maintained between the channels based on the number of actuators and is defined as T/n . The drive electronics were constructed at UNC Charlotte and use a single high-voltage power supply for multiple output channels. By choosing the width of the impulse carefully, the dynamic effects in the output signal can be significantly attenuated.³ The impulse input signal disturbs a broad frequency content which results in a resonant ringing effect. This ringing can be observed between the impulses in the output signal at a frequency equal to the first natural frequency of the system. Theoretical simulations and experimental testing have shown that a pulse width adjusted to be equal to the time period of the first natural frequency of the flexure system suppresses the ringing effect almost completely. This theory can be used to generate a nearly perfect output signal both in simulation and in practice. Much of this theory has been developed for use in dynamic/modal testing where the excitation force source is an impact with duration selected such to supply sufficient energy content to excite the numerous resonant modes of vibration.⁴

* Patent Applied For

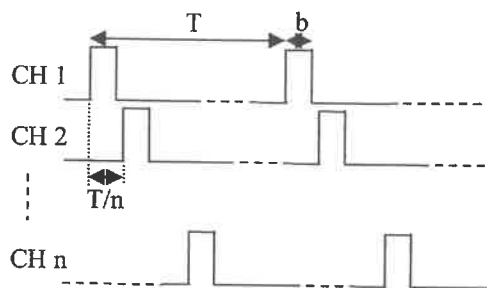


Figure 1: Impulse Driving Signal Representation

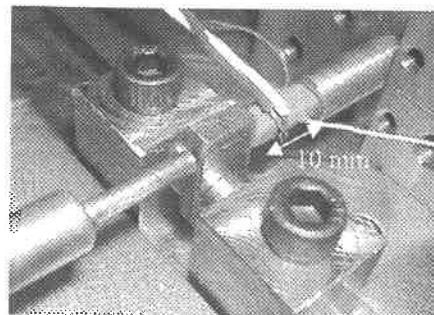


Figure 2: Prototype Torsion Flexure

Prototype Design and Testing

For prototype development, a flexure, shown in Figure 2, was developed that allows for elastic deformation with the objective of observing tool motion at the drive frequency when the flexure is driven below the first natural frequency of the system. Under these conditions there is little phase shift between the command and the response. Numerous flexure systems have been designed, constructed, and experimentally tested using the series actuator drive methodology with satisfactory results. The prototype systems exploited the property of flexures actuated in torsion as having a lower inertia in comparison to translational flexures. The progression in design led to hollow; torsion-based flexures to further reduce the vibrating mass. The piezoelectric stack actuators used in the supporting experiments are Tokin model number AE0203D08 multilayer piezoelectric actuators with an overall length of ten millimeters. For dynamic measurements the bandwidth of the displacement sensor must allow for measurements up to forty kilohertz. An Opto-Acoustics optical sensor with an integral adjustable amplifier was chosen to examine the operating characteristics of the prototype. The preload on the piezoelectric stacks is set by a fine-pitch set screw and flexure deflection.

Typically dynamic measurements are carried out with impact testing techniques. Based on measurement experience, the bandwidth seems to be near twenty kilohertz even with a reasonably small impact hammer with a stiff tip, and it is difficult to supply sufficient energy content to a small area to excite high-frequency vibration modes.⁴ To analyze the dynamic behavior of these relatively small, stiff, high-frequency flexures, two techniques were developed using the piezoelectric actuator as the excitation source, one being a time-domain analysis and the other a frequency domain analysis. The results of the modal testing in the time domain are given in Figure 3. In this case the drive frequency is three kilohertz and the natural frequency of the flexure is forty kilohertz, denoted by the tallest sharp peaks in Figure 4. Note that the pulse width has been adjusted in these experiments such that there is significant resonant ringing in the response and tuning would attenuate this effect. It has been shown experimentally that the intermediate peaks belong to the drive dynamics and not the flexure system.

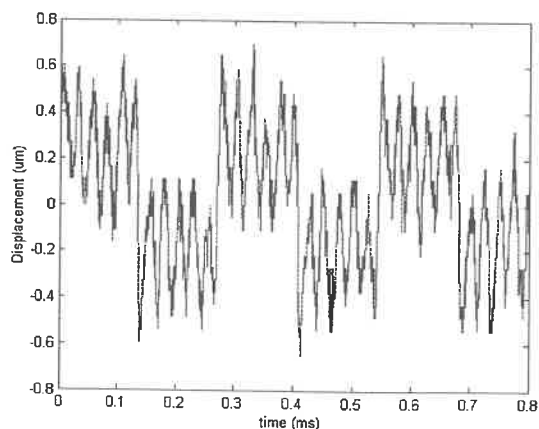


Figure 3: Displacement vs. Time

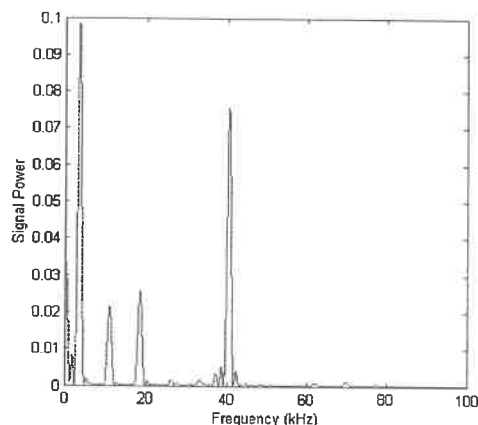


Figure 4: Frequency content of Displacement

The torsion flexure system provided a maximum displacement of 3.5 micrometers at forty kilohertz. The difficulty with the torsion based flexures is the ability to incorporate a cutting edge without significantly mass-loading the system and lowering the natural frequency. Also, the design of the torsion flexure presented here is complex and not

easily altered since there are three primary modes of vibration; the cantilever arm, the shaft translation, and the shaft rotation. A new prototype was designed, built and tested under the same design constraints. A solid steel cantilever beam flexure has been tested with two piezoelectric actuators in the series arrangement. The drawback to the series arrangement is that the actuator next to the flexure only moves the mass of the flexure but the actuator(s) further in the series acts on the flexure plus the mass of the previous actuator illustrated in Figure 5, resulting in a reduction in performance in some cases. One method to eliminate mass loading the actuators is to place the actuators in a parallel arrangement. In the parallel configuration each actuator operates on an equivalent fixed load as demonstrated in Figure 6. The preliminary experimental results for this system verified the theoretical calculations for frequency and displacement for both the series and parallel assembly using the impulse driving technique.

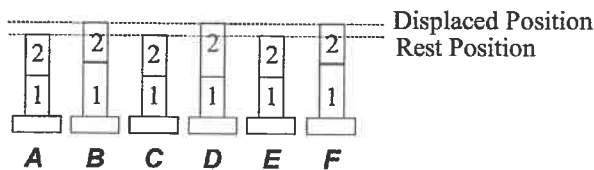


Figure 5: Series Actuation Method

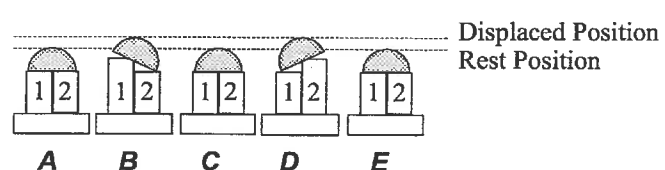


Figure 6: Parallel Actuation Method

Cantilever Cutting Tool

From the proof of concept, the most current design was developed for implementation. This design uses a notched flexure/cantilever beam arrangement as the cutting tool powered by three piezoelectric devices. This design also benefits from the reduction of inertia due to rotary motion versus the traditional translational degree of freedom. In this case the notch flexure defines the center of rotation for the deflection near the clamping interface.⁵ The three stack actuators are operated by phase-shifted impulse signals to achieve a vibratory motion at the tool tip.

Figure 7 demonstrates the principle of the cantilever beam and notch flexure cutting tool during the cutting process. In this case the tool path is defined by the radius from the axis of the notch flexure to the tool tip and the amplitude is a function of the vertical displacement induced from the actuation force denoted by R and a respectively. Due to the impulse driving technique the displacement is unidirectional upward vertical with no overshoot from the unstressed condition downward. With this configuration the actuation force opposes the cutting force and the forces from the cutting process act to maintain the tool to actuator contact interface.

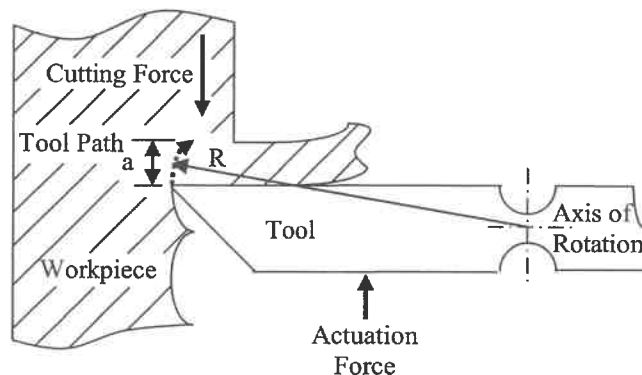


Figure 7: Principle of Cantilever Cutting Tool

Figure 8 shows the assembled cutting tool and three ten millimeter stack actuators during the preload adjustment. The actuators are assembled between carbide half-spheres in an effort to equalize the open-loop flexure output amplitude by providing a single contact interface with the flexure and each of the three actuators. The capacitance gage is used to measure the static preload deflection during the initial set-up. The tool material is micro-grain carbide and was made in the UNC Charlotte machining facilities with a wire EDM before being shipped to Chardon tool for the addition of the diamond tip. The tool nose radius is nominally one millimeter. Figure 9 presents the assembled ultrasonic, vibration-assisted cutting instrument with the optical displacement sensor incorporated to monitor in process activity.

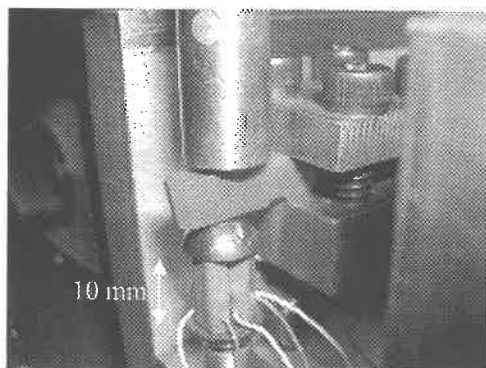


Figure 8: Cantilever, Notch Flexure Tool



Figure 9: Vibration-Assisted Cutting Instrument

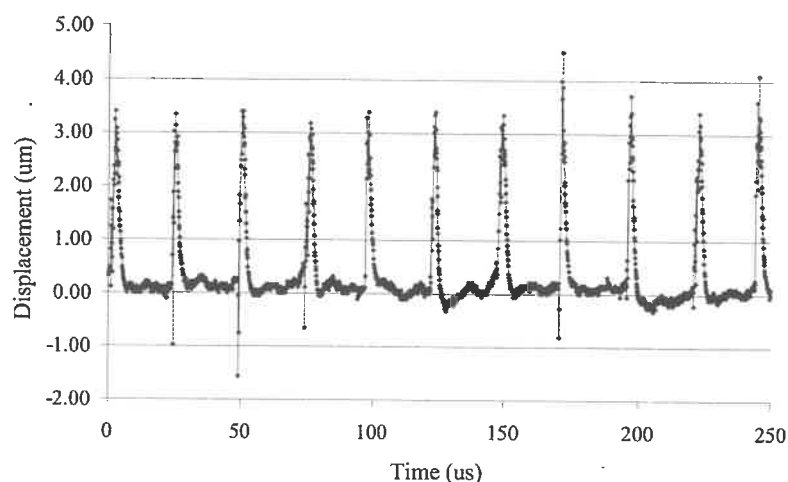


Figure 10: Displacement Data at 40 kHz

The dynamic analysis proves that the flexure will respond up to at least forty kilohertz as desired with the provision for variability in amplitude and frequency. The data presented in Figure 10 shows a mean displacement of approximately 3.5 micrometers as measured between the notch pivot and the tool tip with an input voltage of sixty volts to the pulse generator. In this case, the system consisted of three piezoelectric actuators each operating at 13.33 kilohertz. Based on the pulse generator dynamic characterization, a power supply voltage of sixty volts corresponds to a peak voltage near one hundred volts delivered to each actuator. There are significant fluctuations in the output amplitude which may or may not influence the surface finish of the workpiece since the tool vibrates in a non-sensitive direction for form error. To overcome the amplitude variation a closed-loop control system or lock-in method may be introduced.

Further research includes integration of the tool with a diamond turning machine to produce measurable results on the influence of vibration-assisted machining on workpiece parameters. This includes SEM measurement of the tool geometry for wear and interferometric surface measurements of the workpiece for quality as well as in-situ dynamic force and displacement measurements. These experiments are designed to provide engineering insight into the physical and chemical aspects of the ultrasonic vibration manufacturing process.

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⁵ S.T. Smith, *Flexures: Elements of Elastic Mechanisms*, Gordon and Breach Science Publishers, 2000.

Electrical and Optical Detection and Heating of the High Pressure Metallic Phase of Silicon In-situ During Scratching with Diamond and its effect on the material's hardness

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Keywords: High Pressure Phase Transformation, Silicon, Electrical and Optical measurements, in-situ, scratching

Abstract

Scratching experiments were carried out on silicon wafers (100) with diamond styli of nominal 2, 5, and 10 μm radius. Relevant parameters varied and investigated were: load (10 to 100 mN) and depth of penetration, material hardness, electrical current effects and measured resistance, and speed effects. Direct electrical heating of the metallic high pressure phase of silicon during scratching experiments with currents ranging from 1 mA to 1 A (for resistance heating the material) resulted in substantial thermal softening of the metallized silicon. In addition to the in-situ measurements, post process analysis included AFM and SEM imaging and analysis. For currents above 100 mA (and loads of 50 mN and higher), substantial electrical resistance heating took place resulting in estimated temperatures of 600°C and a corresponding dramatic drop in hardness to about one-half (6 GPa) the nominal ambient conditions value. Direct heating and subsequent softening of the high pressure metallic phase of silicon is proposed as a possible manufacturing augmentation mechanism to reduce tool wear and increase productivity for silicon processing. Further testing with infrared-optical heating will be discussed and reviewed.

1. Introduction

The scratching tests simulate the cutting, grinding and polishing processes. Previous work (Hirata, 2002) demonstrated the feasibility of measuring electrical resistance changes during scratching. The basic idea of the current research is to generate heat and in-situ thermal the metallic soften high pressure phase. This appears to be a very promising way to machine brittle materials in the ductile regime. It is a well-known fact (G.M.Pharr, 1992) that semiconductor materials have an increased conductivity when undergoing a high pressure phase transformation to a metallic state. If a current passes through the metallized material, there will be heat generated within the material (I^2R). As the current increases from micro-amps to milli-amps and even to 1 amp, the heat generated will increase and thus the temperature will rise. With the heat generated by the electrical current, the silicon can be softened and a lower hardness of the material can be obtained. The lower hardness will subsequently result in less tool wear during machining.

In accordance with the above idea, two experiments have been carried out. They are both contact preloaded scratching tests. Scratching is a close simulation to the real cutting, grinding and polishing processes. Unlike one-dimensional indentation, scratching is two dimensional. The first experiment is called scratching with variable currents and the second is infrared detecting and heating system. In the first experiment, a wide range of currents have been tried during scratching with diamond tips. The depth and width of the scratching groove at different current values have been measured and compared to determine the thermal softening effect. The second experiment uses infrared source to detect the high pressure phase transformation and to potentially generate heat to soften the material, i.e. IR laser.

2. Scratching with variable currents

2.1 Experimental setup

Scratching tests are carried out on a silicon wafer with gold patterns on the surface. The silicon wafer is single crystal n type with orientation of (100). A layer of chromium and gold pad with thickness of 3000 angstroms is vacuum deposited. The gold is used for the electrical connection only. Scratching direction is [011]. The stylus is mounted onto a profilometer. Forward scratching speed is 0.305mm/sec and backward speed is 1.5 mm/sec. Scratching length is about 700 μm . Contact preload ranges from 10mN to 100mN. The electrical resistance is measured across the scratch (gap), during scratching. Diamond tips used are of nominal 2, 5, and 10 μm radius. Figure 1A is the diagram of scratching movement.

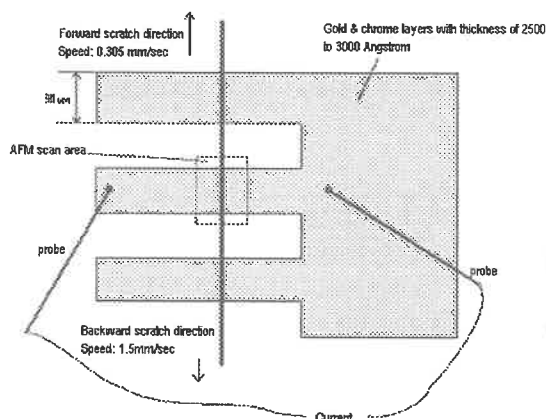


Figure 1A: Diagram of scratching test

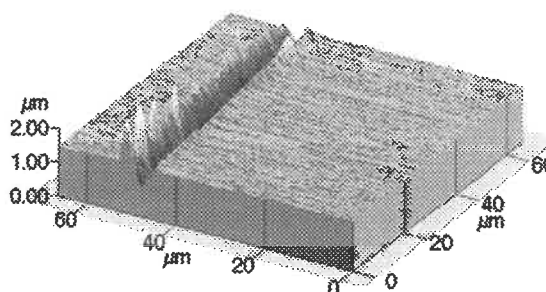


Figure 1B: AFM Image of scratch

2.2 Experiment results

Weights from 10mN to 100mN have been used for the scratching test. We found that a suitable result obtained by using a 5 μm radius diamond tip and 5 to 7 gram weight. A smaller tip is very easy to break, while a larger tip needs many scratches to remove the gold layer. Two different circuits have been tried. One of them provides a constant current and the other provides a constant voltage between the two probes.

Test one: constant current circuit

A circuit is designed to provide constant current with the range from 1 micro ampere to 1 ampere. Scratching groove sizes are measured on AFM for different current values. The results, as shown in figure 2, indicate a trend that the higher the current, the larger the groove depth and width. A plausible explanation for this effect is that electrical heat softens the silicon resulting in plastic deformation.

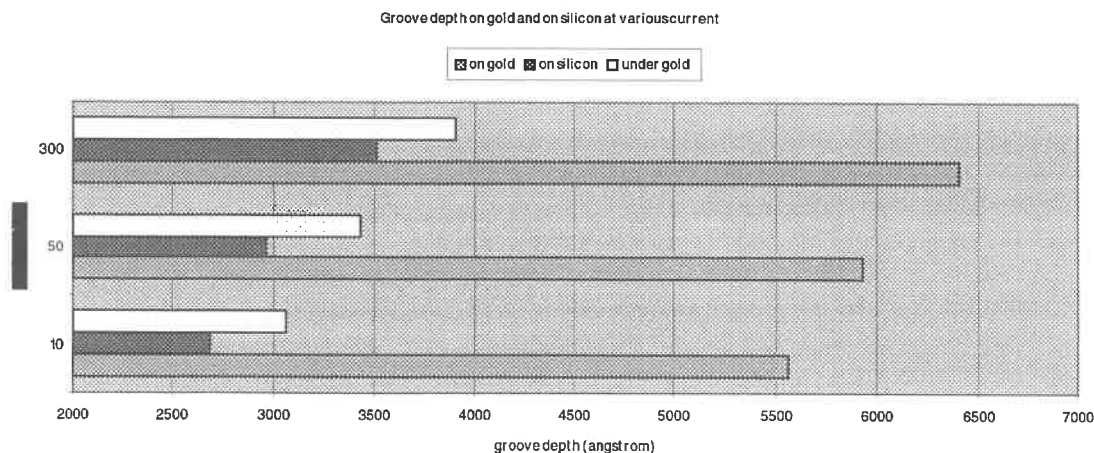


Figure 2: Depth of scratch, on gold and silicon, at various currents (5 gram load with total of 15 scratches)

Test two: V-I circuit (constant voltage)

A constant voltage is applied and a much higher current (approx. 0.65 amps, compared to the max current of 300 mA in the above experiment) is obtained compared to test one, with the result as shown in figure 3. In both test one and two, the probes are set only at the middle gold finger. Therefore current is mainly going through the middle finger. As we can see from figure 3, on the middle finger we have the largest depth and width of the scratch, suggesting that the current produced greater deformation, presumably due to the thermal heating and softening.

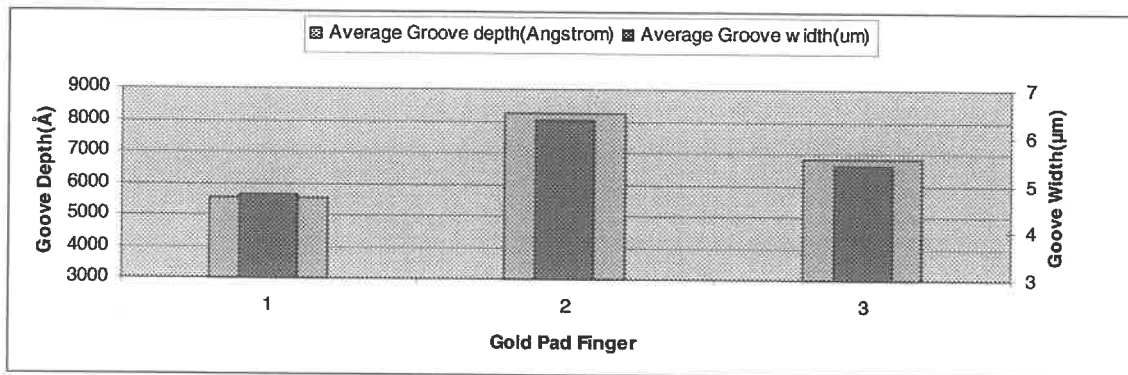


Figure 3: Scratch groove depth and width, with and without current (5 gram load with total of 15 scratches)

3. Infrared detecting and heating system

3.1 Infrared Detecting System setup

A 10 μm diamond tip is attached to the end of a fiber infrared laser (wavelength of 1300 nm, 8mW power), as shown in figure 4. The fiber (2.5mm diameter) with diamond tip is mounted on a profilometer to generate the scratches. As diamond is transparent to infrared light, the laser's infrared light from the fiber end, passes through the diamond tip and into the wafer. With preloaded weight, the diamond tip scratches on pure silicon wafer with crystal orientation (111). Under the wafer there is a 3mm diameter InGaAs detector to measure the transmitted signal. The experimental set up is shown in figure 5.

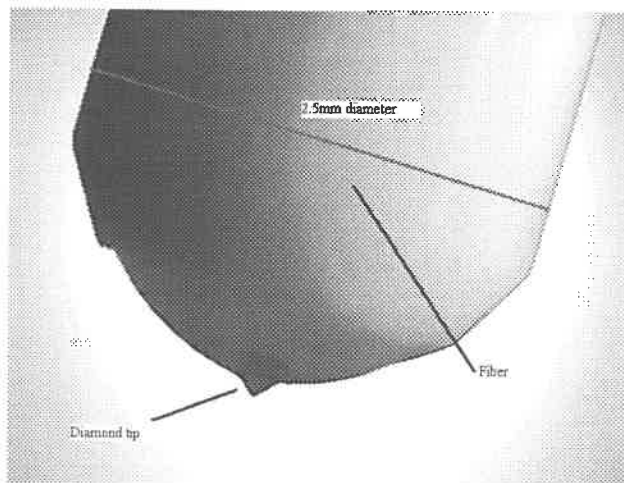


Figure 4 Fiber end with diamond tip (before coating)

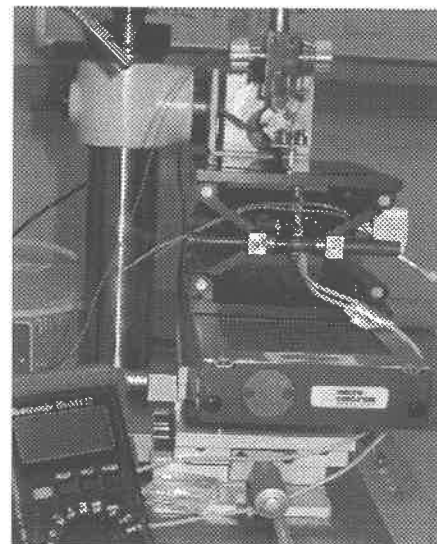


Figure 5: System set up

3.2 Experiment results

A layer (3000 Angstrom) of gold is evenly vacuum deposited on the surface of the fiber end to block the infrared light. When the tip starts scratching the gold on the tip is removed. This insures the infrared light only emits from the very end of the tip. If during scratching a metallic phase is generated, then it will block some of the infrared light and we can detect this change as metals are not transparent in the IR. The scratching length is about 4 mm, which spans across the 3mm diameter detector. Voltage readings with and without preload are compared. Loaded scratching has a lower voltage value, which is proportional to the transmitted IR. Unlike the electrical heating experiment, this test is for a single scratch at a speed of 0.1mm/sec. Figure 6 shows the results with 40mN load. The data clearly indicates a change in the transmitted IR (reduction) due to the applied load, possibly due to the lower transmittance of metallized silicon.

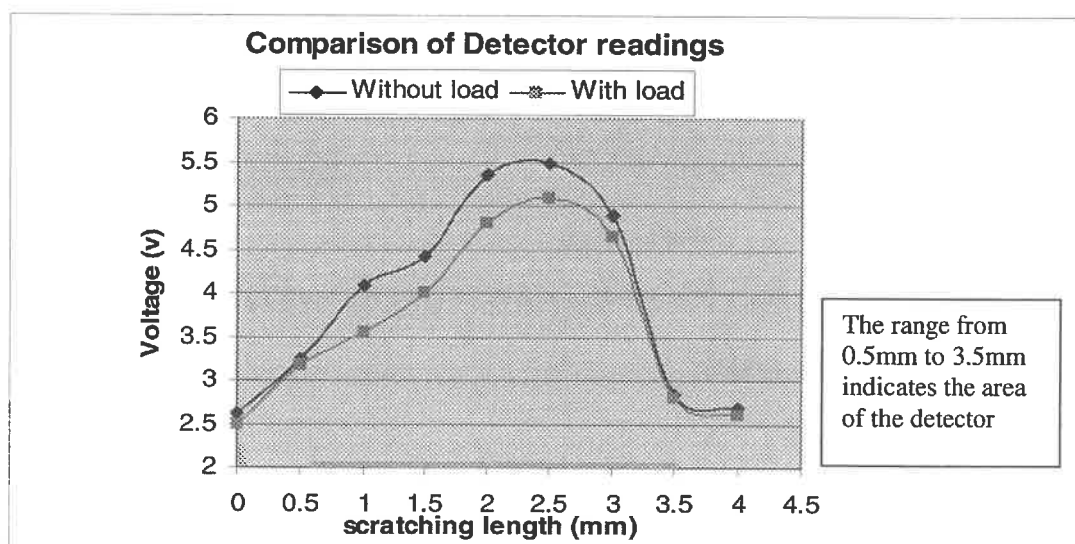


Figure 6: Scratching with and without preload

3.3 Future work

Based upon the success of detecting the high pressure (metallic) phase transformation in-situ via infrared, the next step is to design a laser system to heat the material. If this can be done, it should be a more practical way to carry out ductile-regime machining.

Reference

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CHUCKING PARTS FOR ULTRA-PRECISION DIAMOND TURNING

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Acknowledgment : This paper describes a work made by the people from the Advanced Metrology Program at NIST and Eric Sivignon from INSA, a French engineering school in Lyon, who was completing an internship as a part of the requirements for his degree. It is a product and an example of the cooperation between INSA and the Manufacturing Metrology Division of NIST which receives many future and young graduates from INSA.

INTRODUCTION

This paper reports the different steps used in the design and fabrication of a vacuum chuck which more importantly leads to an innovative and general way of chucking or clamping "parts" in a diamond turning machine tool for ultra precision turning.

A project of the Advanced Optics Metrology Program of NIST is precision figure and thickness measurements of chucked silicon wafers. To make figure measurements to fractions of a nanometer, the wafers are clamped to the chuck with vacuum. Even for wafers up to 300mm diameter, the chuck surface must be plane to well less than 1 micron (peak-to-valley value, P-V). Our chucks are made of mill grade aluminum, 7075 T-657, which are finished using single point diamond turning for a flat front surface. Diamond turning is used to achieve both a good surface finish and especially for the flatness. We found the principal obstacle to making useful chucks was the deformation from chucking the blank onto the turning machine spindle.

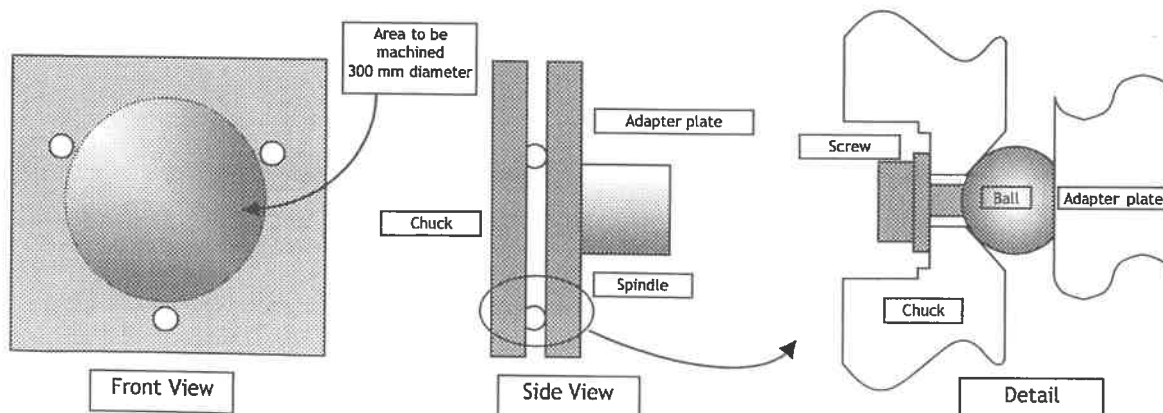
Two successive designs were used to match the requirements, and reach the 0.8 microns flatness. Both are presented in this paper, which starts by explaining the problems with the first design. It also covers the three features of the improved design, 3 points kinematic mounting, spring controlled clamping force, and careful balance.

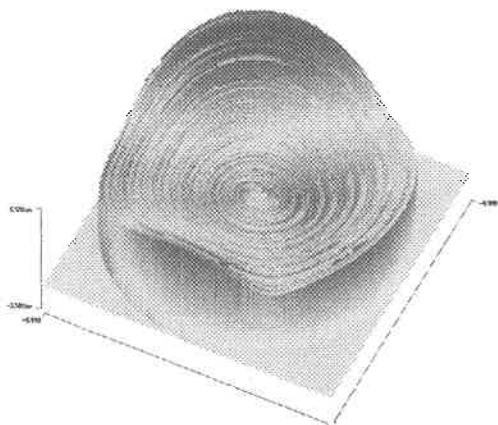
INITIAL DESIGN

The initial design was composed of two rectangular, $\frac{3}{4}$ " thick, aluminum plates: one is the vacuum chuck itself on which the 300 mm diameter flat face must be turned, and the other is an adapter plate which links the spindle to the chuck. This interface was necessary to avoid drilling holes in the flat face because the maximum mounting diameter of the spindle is about 200 mm; the adapter plate is separately screwed to the spindle.

→ 3-points-of-contact kinematic mount

The link between the adapter plate and the chuck is the key feature of the initial design. It is made with a 3-points-of-contact kinematic mount: the three balls are screwed to the adapter plate and the chuck is screwed onto the balls, which seat into three machined vees. The vees are every 120 deg. and their directions are concentric. This design insures that the plate sits on a unique plane, defined by three points. With this kind of mounting, the plate is in an isostatic state, all three translations and three rotations are blocked.





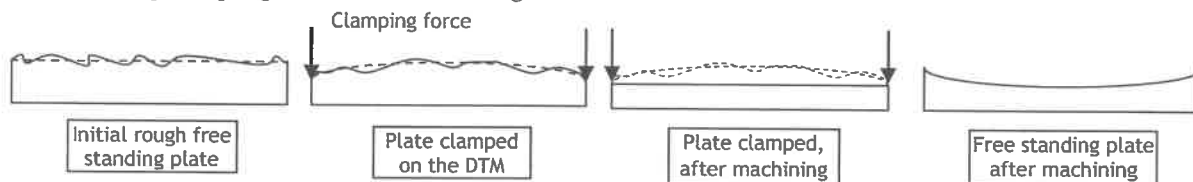
For the first machining try, the three screws were tightened with a torque wrench to 20 lbs.inch (2.26 N.m), the usual value for such an applications.

The NIST XCALIBIR interferometer will be used to measure the wafers flatness was be used to measure the flatness of the chuck: Figure 1 represents a 3D model of the 300 mm diameter face; the peak-to-valley value is over 8 micrometers and there are three "waves" distributed every 120 degrees around the surface.

Figure 1. Optical image of the diamond turned face of a 300mm Diameter vacuum chuck.

IMPROVEMENTS

Using Finite Element Analysis (FEA) to simulate the clamping of the plate in the DTM, the deformation appeared to come from an exaggeration of the clamping force, which deforms the plate before machining; and following machining the plate springs back to a free standing state.



FEA also shows that deformations decrease almost linearly with the clamping force. So the main improvement made here was to reduce as much as possible the clamping force. This includes determining: what was the clamping force in the first case, how little force we can use safely, and how to control precisely the force applying to the chuck.

→ Spring controlled clamping force

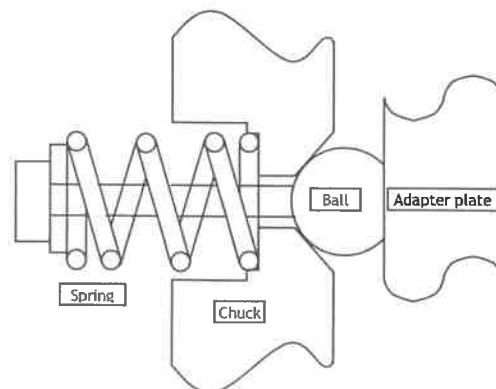
Initially, the three screws were tightened to 20 lbs. inch (2.26 N.m). Considering the friction coefficient of a steel/aluminum interface, the axial force corresponding to the torque and the type of screw is about 450 lbs (2000 N) on each screw. The axial force per bolt to transmit the rotation from the spindle using the only friction between the balls and the plate (without any shearing on the screws) is 100 lbs, or 450 N. The only radial load is the weight of the chuck. To have a good idea of the clamping force, die springs are now used. For so small forces, it was hard to control the torque on the screws, and the uncertainty was very high using a torque wrench. With a constant stiffness spring, the displacement is immediately related to the axial force.

The springs used in this case have the following properties:

- rate : 1080 lbs./inch
- diameter : $\frac{3}{4}$ "
- length : 1"
- wire size : 0.125"

The deflection corresponding to a 100 lbs. load is 0.093"

Controlling precisely the clamping force made possible to reaching a P-V value of 1.9 +/- 0.05 micrometers over the full 300 mm aperture after re-machining.



→ Balanced design

But now the plate error had a different shape; which led to focusing on the imbalance while machining on the DTM. According to the last measurements, there is still a bending of the chuck but with a shape other than the simple 3 waves. By turning again the chuck with an angular offset between the adapter plate and the chuck the use of an adapter plate was disturbing the chuck.

The following Figures 2 and 3 show the shape of the deformation is directly related to the direction of the adapter plate while machining ; the non-axisymetry creates a weak point in the assemble. To avoid these deformations, there were two possible solutions: the first was to make the adapter plate stiffer by either changing the material or increasing the thickness; the second was to clamp the chuck directly onto the spindle of the DTM. The first solution would increase the weight supported by the spindle, which is not an especially good idea. The second requires drilling holes in the chuck face, but optical measurements could still be completed. So the choice was made to fabricate a new chuck that could be bolted directly on the spindle of the DTM.

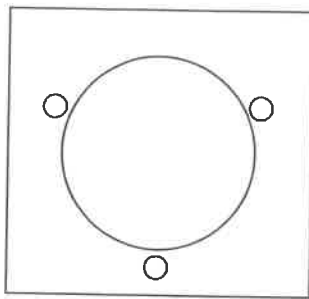


Figure 2. The Adapter plate and chuck with the same orientation for the machining

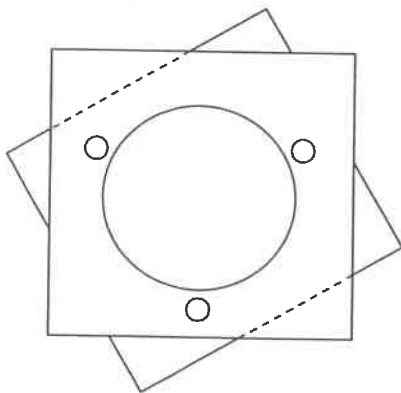
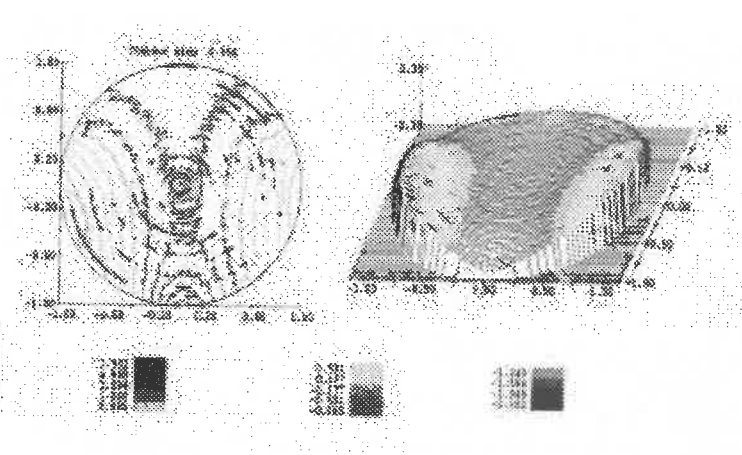
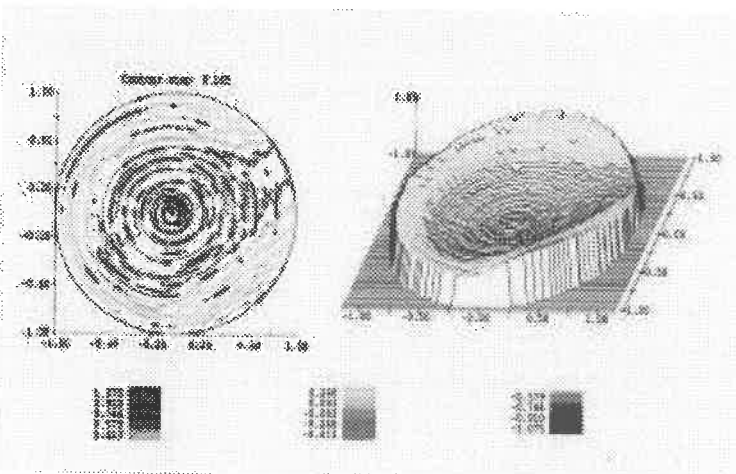


Figure 3. Adapter plate and chuck at 120 deg.



NEW DESIGN

The new design takes advantage of the three major points already developed: the plate is now a 2" thick round aluminum plate that can be bolted onto three balls screwed to the spindle. This thickness was necessary to include the springs in the chuck, to be able to turn the face. With this way of mounting the vacuum chuck, the precision reached is a P-V value of 0.8 ± 0.05 microns. See Figure 4.

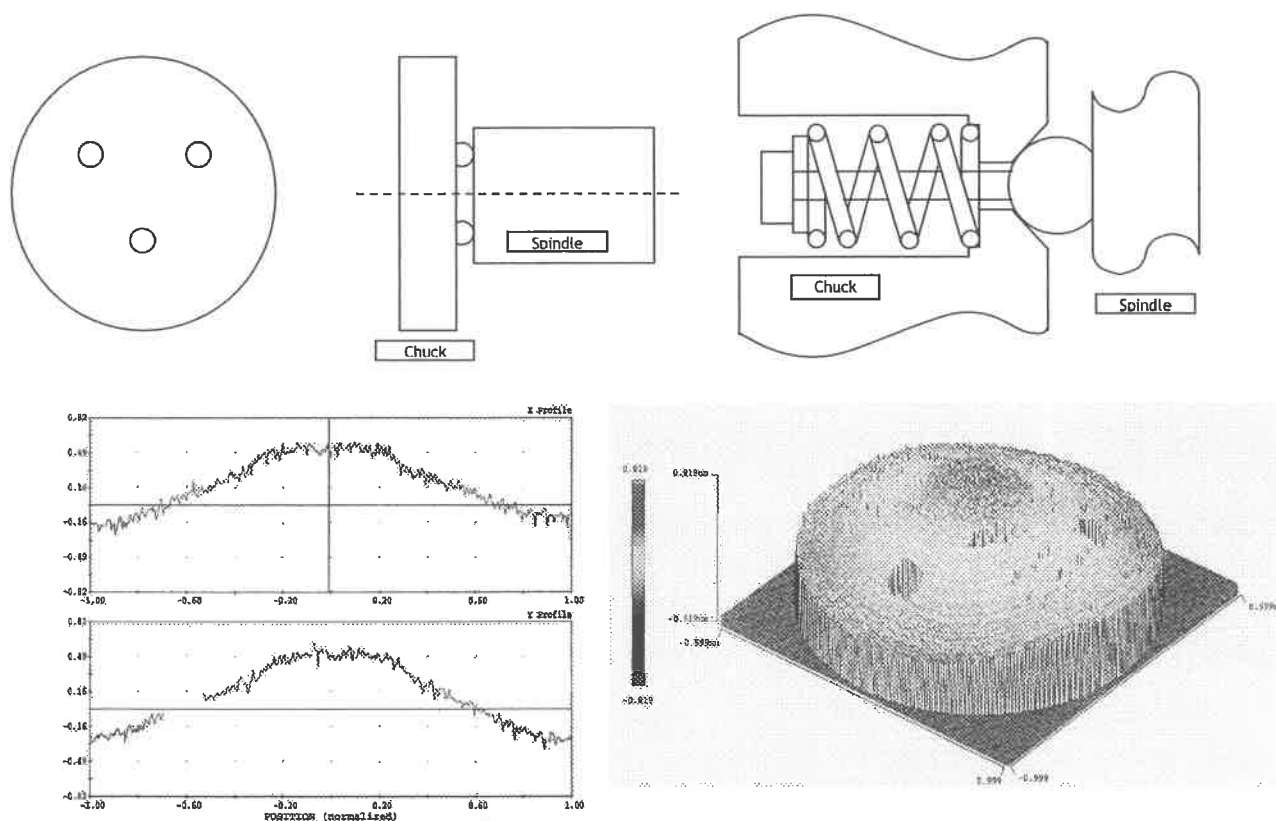


Figure 4. A schematic of the mounting of the chuck and below an interferogram of the resulting Chuck flatness.

The 0.8 micrometer chuck was measured mechanically while in the diamond turning machine and optically with the NIST XCALIBIR interferometer. Both measurements show the same conical shape. We think this means, we have reached the capabilities of the diamond turning machine itself. The machine is certified for a deviation of 0.7 microns due to the misalignment between the X axis and Z over full travel.

CONCLUSION

More than the simple achievement of a very flat vacuum chuck, this work also gave a useful view of the behavior of parts mounted of the diamond turning machine. Depending on the precision desired, it is now obvious that every detail is important; a kinematic mount will reduce the likely bending of the part, but the clamping force must carefully be adjusted and be a well adapted blank geometry can lead to even better results.

Ductile mode cutting with torsional vibration for glass material

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1. Introduction

Machining of hard and brittle material, especially glass is still a major problem because of its lower fracture toughness and higher hardness. Glass tends to crack easily during machining under small stress. Many researchers tried to develop new methods for machining of brittle and hard material. Vibration cutting was one of the new methods for this purpose. In early experiments of vibration cutting, the tool was vibrated in the straight line along the cutting path [1]. Shamoto et al. [2] have reported more efficient method by elliptical vibration cutting. By the way, the hard and brittle material can be machined as a ductile material under the sufficiently small depth of cut [3]. However, increasing the critical depth of cut of glass is still difficult. Moreover, the severe wear of a diamond tool is also the serious problem because the tool wear not only raises the machining cost, but also degrades the machined surface of glass.

In this paper, a new cutting method with torsional vibration was proposed to increase the efficiency of material removal rate for glass material. The experiment was conducted in order to investigate the brittle-ductile characteristics of glass cutting and torsional vibration cutting phenomena.

2. Experimental setup

In the experiment, a soda-lime glass was selected as the workpiece material. The cutting tool is a single crystal diamond with a nose radius of 1.5 mm and a rake angle of -15° . The experiment was conducted on shaper machine, which was developed by the authors [5]. The workpiece was set with the inclination rate of $5 \mu\text{m}/30\text{mm}$. Therefore, the depth of cut increased gradually in the feed direction. The feed speed was 1 mm/s. The torsional vibration was applied to the diamond tool with 125 Hz of vibration frequency and 0.63 degree of torsional angle amplitude.

A photograph of the experiment setup is shown in Fig.1. The major components of the developed system consist of a tool feed table driven by linear motor, a three-axis force sensor (model type 9256A1, Kistler Ltd.), a tool positioning unit, a workpiece alignment unit and a torsional vibration unit.

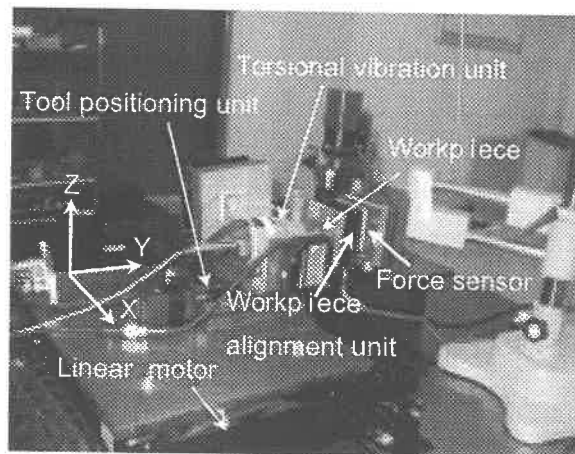


Figure 1 Experimental setup

Figure 2 shows the torsional vibration unit with four piezoelectric actuators that were connected directly to the tool holder. When two sinusoidal voltages, which were 180 degrees out of phase were applied to two pairs of PZT (a), PZT (b) and PZT(c), PZT (d) respectively (Fig.2 (c)), the cutting tool was vibrated torsionally as shown in Fig.2 (b).

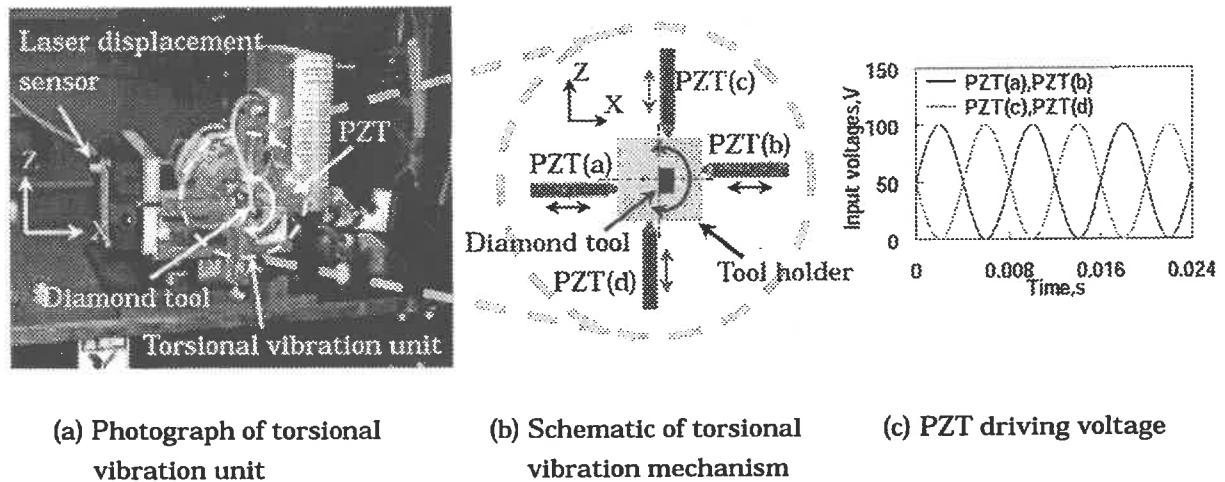


Figure 2 Torsional vibration unit

3. Experimental results

The groove cutting experiments using the developed shaper machine were carried out in order to investigate the brittle-ductile characteristics of glass cutting. Accordingly, the comparison between the torsional vibration and nonvibration cutting phenomena was conducted with the same cutting conditions shown in Table 1. The experimental results were shown in Figs.3 and 4.

Table 1 Cutting conditions

Cutting tool	Single crystal diamond (R1.5 mm, Rake angle -15 degree)
Workpiece	Soda-lime glass (30x30x5t)
Feed speed	1 mm/s
Workpiece inclination rate	5 μ m/30mm
Torsional vibration	Frequency: 125 Hz Amplitude: ± 0.63 degree

Figure 3 and 4 show the photograph of groove, cross section profile and cutting forces generated in orthogonal cutting and torsional vibration cutting respectively.

In Fig.3 (a), the groove was smooth at the beginning, and then changed to be rougher with crack propagation near section a-a. Nevertheless, in Fig.4 (a) crack was not observed even cross section b-b, which was measured at 2 mm from the edge of workpiece.

The critical depth of cut that was measured near the transition boundary (cross section a-a) shown in Fig.3 (b) was 0.67 μ m by orthogonal cutting. In Fig.4 (b), as the torsional vibration was added with the same cutting condition, the depth of cut of 3.46 μ m for ductile mode cutting was achieved.

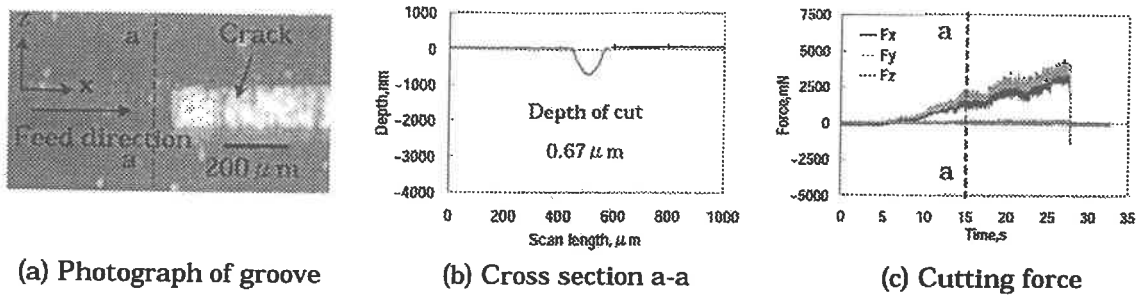


Figure 3 Experimental results for orthogonal cutting

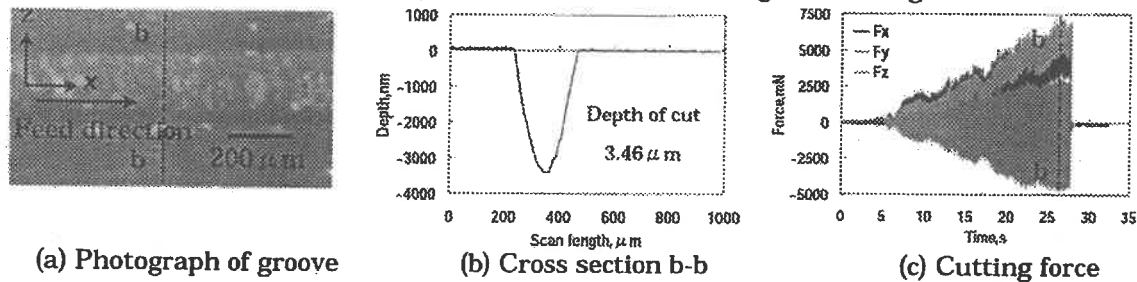


Figure 4 Experimental results for torsional vibration cutting

In Fig.3 (c) and Fig.4 (c), as the depth of cut increased, F_x and F_y became larger. However, F_z was approximately 0 N in orthogonal cutting, while F_z was observed in torsional vibration cutting.

4. Discussions

In Fig.3, the critical depth of cut was less than $1 \mu\text{m}$. F_z did not occur. Moreover, crack was observed by orthogonal cutting. From the viewpoint of fracture mechanics, the compressive stresses were existed in X and Y directions. Nevertheless, it was less in Z direction. Therefore, mode II (shearing) or mode III (tearing) of fracture might occur in the vicinity of tool tip as shown in Fig.5 (a).

In Fig.4, the critical depth of cut was larger than $3 \mu\text{m}$. F_z was also generated. Nevertheless, crack was not observed by torsional vibration cutting. The reason for this improvement is hypothesized that the compressive stresses are existed in X, Y and Z directions. They caused the stress intensity factor at crack tip to reduced until it became less than the fracture toughness. Therefore, crack propagation due to pre-existing flaws was suppressed as shown in Fig. 5 (b). This result was consistent with the previous study by Yoshino et al. [4]. He has reported that the critical depth of cut was found to increase significantly in scratching test on brittle material under hydrostatic pressure.

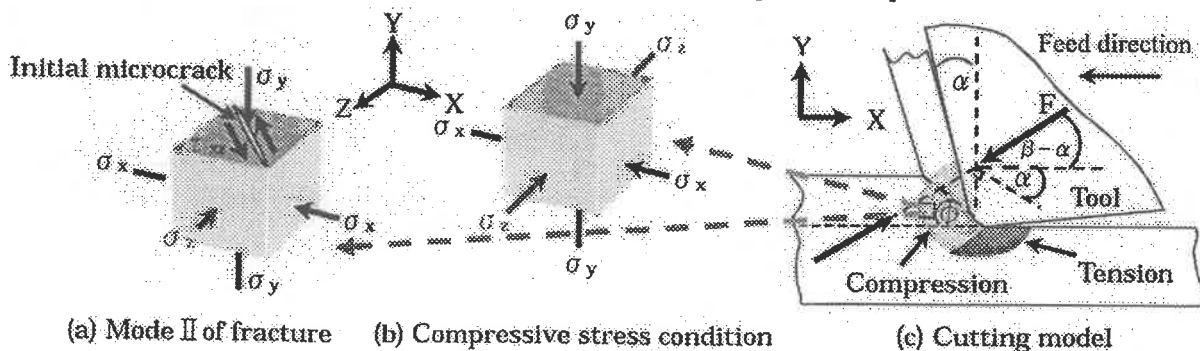


Figure 5 Difference of stress condition between orthogonal and torsional vibration cutting

Then an oblique cutting with 45 degrees of cutting edge inclination angle was conducted to confirm the hypothesis. If F_x , F_y and F_z are generated in cutting process, 3-dimensional compressive stress condition will be able to suppress the crack propagation. The experimental results with the same cutting conditions as the orthogonal cutting experiment were shown in Fig. 6 as follows.

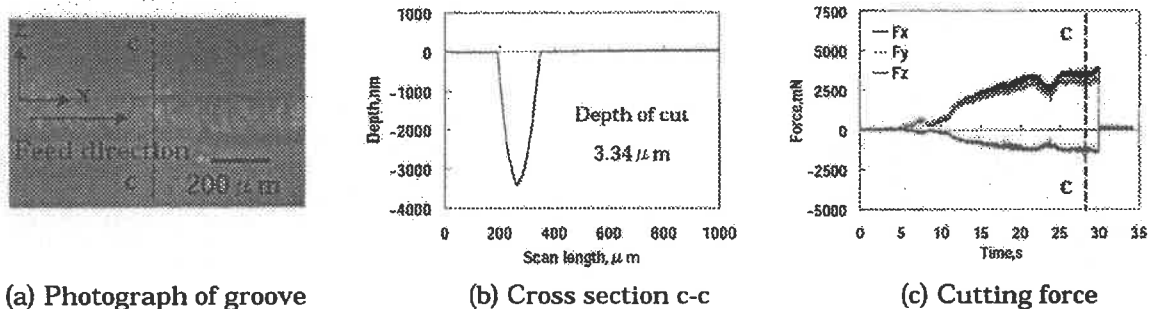


Figure 6 Experimental results for oblique cutting

In Fig. 6, the critical depth of cut was realized larger than $3 \mu\text{m}$. In this case, as shown in Fig. 6 (c), F_x , F_y and F_z were generated. Though there were small subsurface cracks beneath the cutting surface, a large crack did not appear in oblique cutting. These results strongly supported the above mentioned hypothesis that the crack propagation was suppressed in 3-dimensional compressive stress condition generated in the cutting process.

5. Conclusions

The torsional vibration cutting method was proposed to realize ductile mode cutting of glass material. The experimental results show that the proposed method can increase the critical depth of cut for glass material larger than $3 \mu\text{m}$. The main reason for the improvement of critical depth of cut by torsional vibration is hypothesized that 3-dimensional compressive stress condition at the vicinity of tool tip is able to suppress the crack propagation. In addition, the oblique cutting was also conducted to confirm the hypothesis. The experimental results show that the critical depth of cut more than $3 \mu\text{m}$ was also achieved. From these, the hypothesis is strongly assured by the experimental results. However, further experiments and analyses should be conducted in the near future.

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Edge Rounding Method by Means of Water Jet for Cylindrical Rollers of Bearing

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1. Introduction

The improvement of the seizure resistance of bearing is very important for the quality improvement of a bearing. Edge rounding of the roller is one of the most important factors for the improvement. Specifically speaking, there are two connection edges; (i) the connection edge on the cylindrical surface for preventing stress concentration and (ii) the connection edge on the facing surface for preventing scuffing into the raceway surface as shown in Fig.1. In the paper, a new machining method by means of water jet is proposed to efficiently obtain constant and large radius of curvature as much as possible. The paper shows experimental investigation results on the relationship of machining conditions with machined profile and surface roughness.

2. Edge rounding method

Figure 2 shows the edge rounding method for roller bearings by means of water jet. Abrasive water jet collides with the cylindrical roller at small angle with some stand-off distance. The cylindrical roller is fixed on the tip of a spindle and rotates during machining. There are two edges to be rounded on the cylindrical surface side and the facing surface side. In case of the cylindrical surface side, the workpiece and the water jet should be perpendicular to each other. In case of the facing surface side, the workpiece should be fixed with small inclination angle to decrease the interference of the water jet with the facing surface (Fig.3).

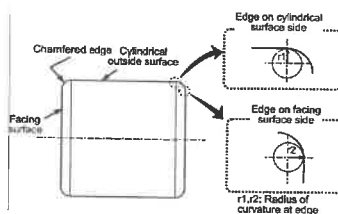


Fig.1 Cross section of a bearing roller

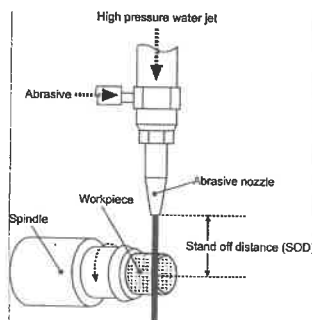


Fig.2 Rounding method

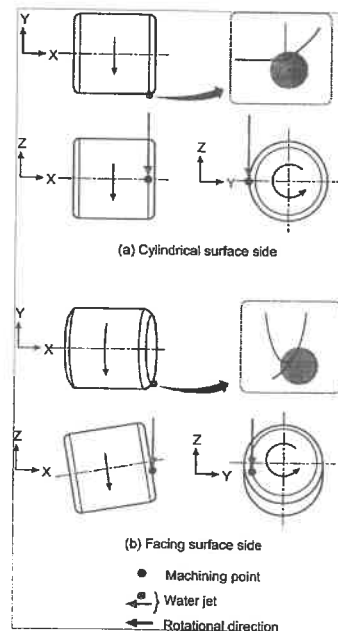


Fig.3 Workpiece orientation

3. Machining model and simulation

Figure 4 explains the machining model on the facing surface side. In the figure, the erosion process of a point on the edge during workpiece rotation is displayed[1]. When a point on the workpiece comes near the outer of the water jet (point A), some abrasives erode the point on the workpiece with stand-off distance of S_A and impact angle of α_A . Then, as the workpiece rotates, the stand-off distance increases and the impact angle decreases. Finally, some abrasives near the center of the water jet erode the point with stand-off distance of S_B and impact angle of α_B . The amount of erosion during a rotation is equal to the amount of erosion during the period mentioned above[2].

Using the machining model mentioned above, a machined surface profile can be simulated to determine the machining conditions, especially the injection point. The simulation is based on the discrete model of the machining[3]. Machining ability distribution within a water jet is represented as the discrete erosion ratio obtained by some groove machining experiments as shown in Fig.5 (a). A workpiece is also modeled discretely as shown in Fig.5 (b). The simulation results as shown in Fig.6 suggest the appropriate injection region exists.

4. Experimental setup and method

The aqua jet pump (AJP-35020: Sugino Machine Co. Ltd.) is used to produce ultra high pressured water. The system consists of two translation axes (X and Y axes) to determine the relative position between a workpiece and nozzle, one orientation axis (B axis) to incline the workpiece with the spindle, and one translation axis (Z axis) to determine the stand-off distance. The influence of water jet pressure, abrasive mesh size and stand-off distance is investigated in the experiments. Machining profile and surface roughness measurements are done by means of a contact-type profilometer (Form Talysurf S3C Rev.01.01 Standard: Rank Taylor Hobson Ltd.) with 2um of probe radius and 70~100mg of contact force. Profile estimation is done by radius of curvature on the machined surface.

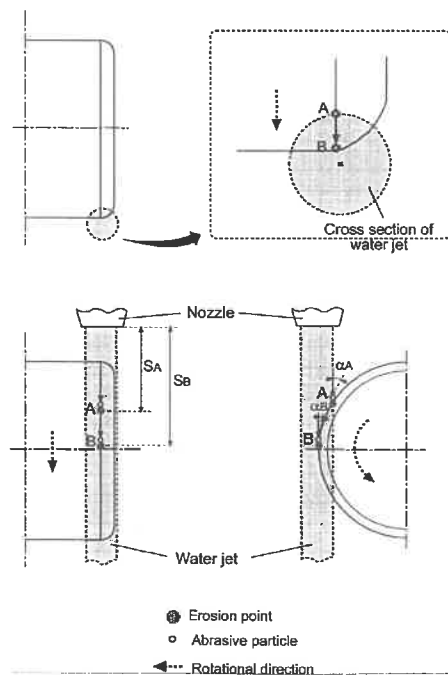
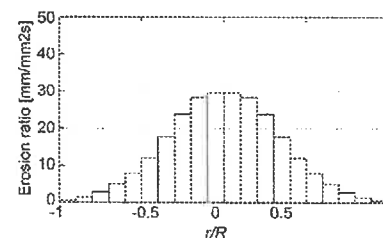
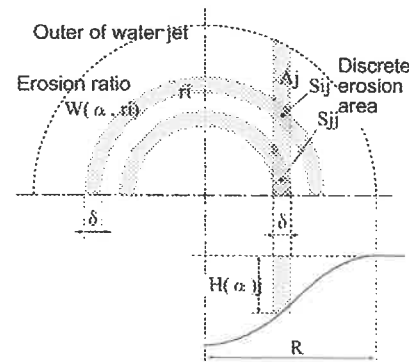


Fig.4 Machining model on cylindrical surface



(a) Discrete erosion ratio within water jet (injection angle $\alpha = 10^\circ$.)



(b) Discrete machining model

Fig.5 Machining model

4. Experimental results

4.1 Comparison of profile between simulation and experiment

Figure 7 shows comparisons of profile between simulation and experiment. They suggest that the simulated profile would represent the experimental result very much. Furthermore, machining time is also very important as well as injection position to affect machining profile very much. From the results, the simulation model is appropriate enough to determine cutting conditions in advance.

4.2 Relationship between machining conditions and surface roughness

The relation between water jet pressure and surface roughness is shown in Fig. 8, and the relation between abrasive mesh size and surface roughness is shown in Fig.9, respectively. It turns out clearly from those figures that the surface roughness is hardly dependent on water jet pressure but the mesh size of abrasive. Furthermore, as compared with cylindrical surface side, the surface roughness of facing surface side shows the good result. In the experiments, the surface roughness of $0.5 \sim 0.7 \mu\text{m Ra}$ on the cylindrical surface and that of $0.4 \sim 0.5 \mu\text{m Ra}$ on the facing surface were realized.

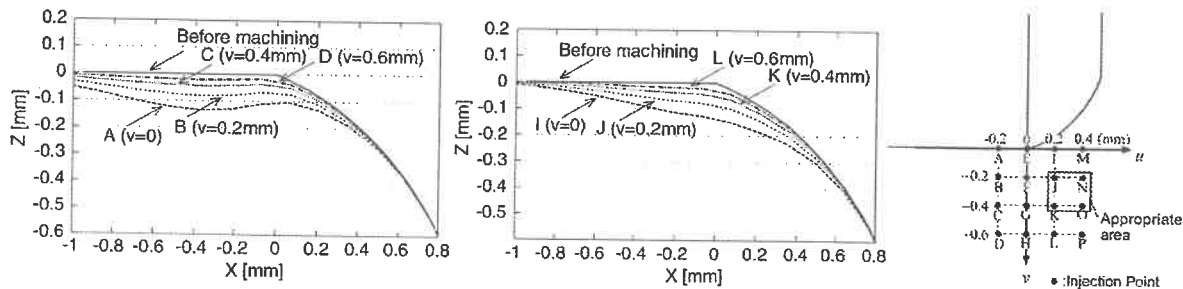
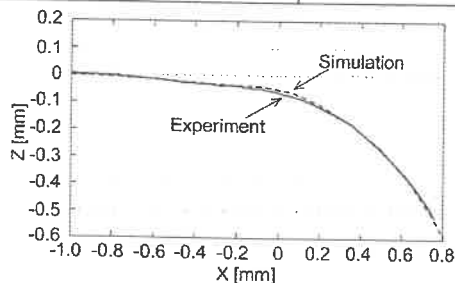


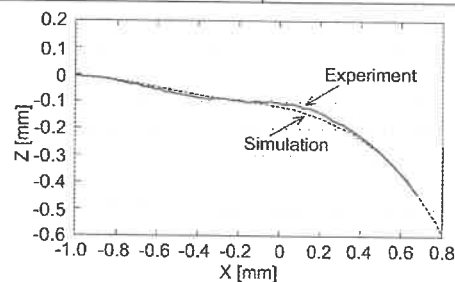
Fig. 6 Simulation results of a machining profile on the cylindrical surface side
(Pressure 150MPa, Mesh #150, SOD10mm, Machining time 10s)

Table 2 Experimental conditions

Water jet pressure [MPa]	100, 150, 200	Stand off distance (SOD) [mm]	10, 20, 30
Spindle rotational speed	300rpm	Machining time [s]	10, 20, 30
Abrasive nozzle diameter	1.5mm	Water nozzle diameter	0.25mm
Abrasive flow rate [g/s]	2.5~6.5	Workpiece material	SUJ2 (JIS)
Abrasive material	SiC	Abrasive mesh size	#80, #150, #220



(a) Machining time 10s



(b) Machining time 30s

Fig. 7 Comparison of profile between simulation and experiment (Pressure 150MPa, Mesh #150, SOD10mm)

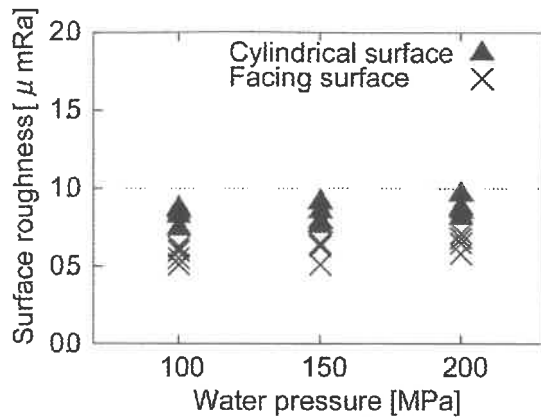


Fig.8 Relationship between waterjet pressure and surface roughness
(SiC#150, SOD 10mm, Machining time 20s)

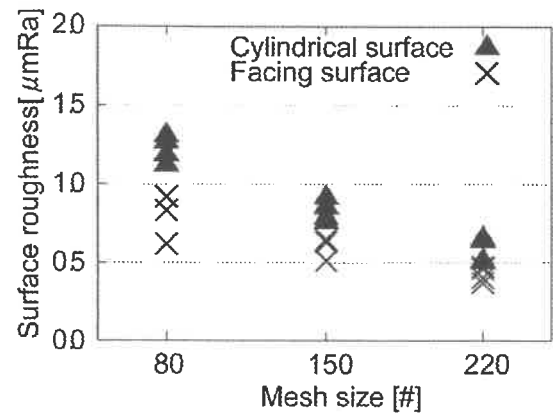


Fig.9 Relationship between abrasive mesh size and surface roughness
(150MPa, SOD 10mm, Machining time 20s)

5. Discussions

It is proved that the proposed machining method in this paper has capability to realize the desired round profile at the edge of a cylindrical bearing roller. One of the requirements for bearing rollers is good surface roughness. It is specifically said that they are required less than 0.3 $\mu\text{m Ra}$. The results didn't realize the value, which is consideration in selection of abrasive mesh size. It is thought that the proposed method will contribute to the improvement in edge quality of not only bearing rollers but extensive variety of mechanical parts.

6. Conclusions

In this research, the new rounding method using water jet by which desired edge profile can be acquired efficiently was proposed. In the proposed method, water jet pressure, abrasive mesh size, stand-off distance and water jet injection angle are taken as machining conditions. The simulation model to generate machined profile is proposed and proved to be appropriate. The model is very indispensable to determine cutting conditions beforehand. Effects of those cutting conditions on machining profile, surface roughness and machining time are examined by several experiments.

Acknowledgement

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Inspection of Micro-Drilling Processes by using the On-Machine vision

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Abstract: In order to inspect burrs and machining quality in micro-drilling processes, a cost-effective method using an image processing and a shape from focus (SFF) methods on the machine tool is proposed. As the on-machine vision units are incorporated with the CNC function of the machine tool, direct measurement and condition monitoring of micro-drilling processes are conducted between drilling processes on the machine tool. Stainless steel and hardened tool steel are used as specimens and twist drills made of carbide are used in experiments. Validity of the developed system is confirmed through experiments.

Keywords: Burr, CCD Camera, Condition Monitoring, Hole Quality, Illumination Unit, On-Machine Measurement, Shape from Focus, Zoom Lens

1. Introduction

Micro-drilling processes have been widely used to produce micro holes such as micro dies and molds, fuel injection nozzles, watches, bearings and printed circuit boards, etc. And it has more attention in a wide spectrum of precision production industries.

The burr (height, width) and hole quality (oversize, location error) have a significant effect on assembly of precision components. Fast and accurate measurements of burr geometry and hole quality are important for the condition monitoring of micro-drilling processes. In order to measure burr and hole quality, measurement systems such as SEM (Scanning Electron Microscope) and confocal microscopes have been used [1-2]. However, these measurement systems are very expensive. They are difficult to apply condition monitoring of drilling holes between micro-drilling processes. In addition, they are not cost-effective.

In this paper, a machine vision technique using the SFF method [3] is applied to measure the micro-drilling burrs and hole quality. In order to obtain clear images and reduce noise due to reflection, a novel illumination unit is devised using the LED array and a halogen lamp. Experiments are conducted with twisted carbide drills, and stainless and hardened tool steel specimens.

2. Hole Quality

Experiments are conducted to investigate the effectiveness of drilling processes by measuring the hole quality after machining. Two aspects, location error and oversize of the hole, of drilling quality are measured. Depending on type of drills and drilling processes, it may occur that the drill walks on the surface of the workpiece before entering the part. Whirling of the drill

edge at the time of penetration into the workpiece degrades hole quality as well.

The drilled center location of a hole is determined by the position where the drill comes to a stop after walking. The location error deviated from the desired center location is called the location error of the hole [4], E_L shown in Fig. 1. Actual diameter is calculated by evaluating the result of least squares technique after scanning the edge through the edge detection and image processing. And location error is measured by the difference between the actual hole center and the desired hole center with comparing the CNC position and image coordination.

In all drilling operations, diameter of the hole is predetermined by the diameter of a drill bit. However, some variation in hole size from the nominal (tool) value is expected to occur even under the best drilling condition because of tool wear and the whirling. The amount of deviation between the actual hole diameter and the nominal diameter is called diameter deviation [4], Δd_e as shown in Fig. 1.

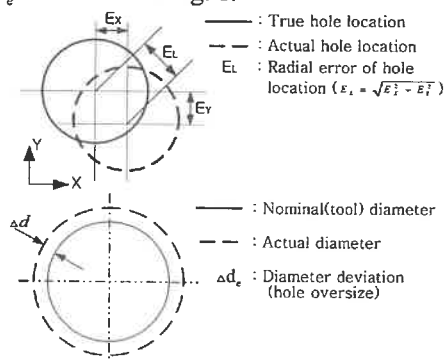


Fig. 1 Location error and diameter deviation.

3. Burr Inspection

3.1 Focus and defocus Images

When the object point P is blurred on the image detector plane ID shown in Fig. 2, the radius r of the blurred circle Q' is obtained. This image $h(x,y)$ is the point spread function (PSF) of the camera. It is also called the blurring function. A two-dimensional Gaussian function is used to approximate this physical model. Then, the blurred or defocused image $g(x,y)$ at the small image region on the image detector ID is equal to the convolution of the focused image $f(x,y)$ and the PSF $h(x,y)$. Hence, if G , F and H are the Fourier transforms of g , f and h , respectively, then $G=HF$. $H(\omega, \nu)$ has characteristics of a low-pass filter where the bandwidth decreases with increase in defocusing. Defocused images of the object can be obtained by moving the lens every Δd with respect to the reference plane as shown in Fig. 3. The increment of the stage movement Δd is manipulated by the CNC function of the machine tool.

3.2 Shape from focus

In Fig. 3, it is assumed that the reference plane of the block gauge corresponds to the reference focus position of the machine tool. If the camera is moved toward object focus position using the z-axis movement of the machine tool, the image will gradually increase its degree of focus (high frequency energy) and will be optimally focused at the object plane of the block gauge. When the object plane has a perfect focus at d , object height d_z is given by

$$d_z = d = N \times \Delta d \quad (1)$$

This approach may be applied independently to all surface elements to obtain the shape of the entire surface.

3.3 Focus measure operator

It is known that the quality of focus is proportional to the amount of high frequency energy [3]. Several focus measure operators have been proposed [5]. In this study, a Laplacian operator is used because it has good performance with a sharp peak.

The Laplacian of two-dimensional images is given by

$$\Delta^2 I(x, y) = \frac{\partial^2 I}{\partial x^2} + \frac{\partial^2 I}{\partial y^2} \quad (2)$$

The Laplacian can be generated by the convolution operation with the 3×3 kernels $H_L(x, y)$ in spatial domains as follows:

$$L(x, y) = I(x, y) \otimes H_L(x, y) \quad (3)$$

where $H_L = \begin{bmatrix} -1 & -1 & -1 \\ -1 & 8 & -1 \\ -1 & -1 & -1 \end{bmatrix}$

The focus measure at a window is computed as the sum of the modified Laplacian values, in a small window around (i, j) , that are greater than a threshold value:

$$f(i, j) = \sum_{x=i-M}^{i+M} \sum_{y=j-N}^{j+N} L(x, y), \text{ for } L(x, y) \geq T_L \quad (4)$$

We shall designate the Eq. (4) to the Sum-of-Laplacian (SL). In Eq. (4), parameters of M and N determine the window size that is used in computing the focus measure. The window size is selected empirically.

3.4 Illumination unit

One of the most important parts of the hardware configuration is the illumination. For burr and hole quality measurements, the main requirement of system is to provide adequate contrast between burr and background. Intensity of illumination source should be adjusted to accentuate the burr region of interest. In addition to the selection of an appropriate light source, consideration must also be given to the technique that will give the optimum result [6].

In this paper, we apply both the front lighting with coaxial halogen light source and back lighting with LED-array illumination to inspect hole quality and burrs.

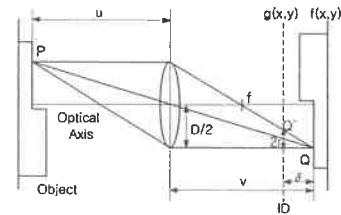


Fig. 2 Image formation of focus and defocus.

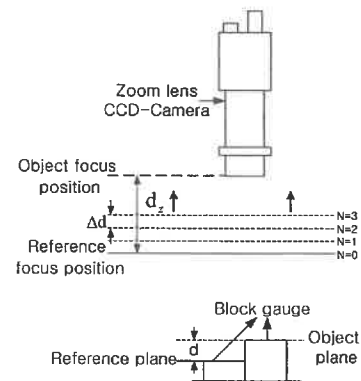


Fig. 3 Shape from focus(SFF).

4. Experiments

4.1 Experimental setup

Measuring system consists of a CCD-camera with zoom lens attached to the vertical CNC machining

center, illumination units of both LED and halogen light source, a digital image processing unit, and a micro-computer as shown in Fig. 4.

The image from CCD camera with zoom has 640×480 pixel elements with a pixel size $1.49 \mu\text{m}$ (H) $\times$ $1.45 \mu\text{m}$ (V). The camera axis is perpendicular to the stage, table of machining center, holding a workpiece. Image acquisition and drilling processes are operated simultaneously by NC program. Images are captured through the CCD camera by a frame grabber with 256 digitized gray levels.

4.2 Calibration

Fig. 3 shows block gauges to calibrate the developed SFF system. First, the difference d of $50.1 \mu\text{m}$ is calibrated by an LVDT with resolution of $0.1 \mu\text{m}$. After selecting Δd to $8 \mu\text{m}$ and window size of 30×30 , SL focus measure function is obtained as shown in Fig. 5. From this figure distances between the reference and the focused object surfaces are 72 and $120 \mu\text{m}$, respectively. Comparing with the calibrated results by LVDT, we can confirm that the developed SFF system has $2.1 \mu\text{m}$ error as shown in Table 1.

However, distance estimations through the SL focus measure function gives local values contaminated by noises. Gaussian interpolation with step size of $0.1 \mu\text{m}$ is applied to find the optimal values of focus measures [3]. Fig. 6 shows optimal Gaussian interpolation results of the focus measures.

5. Results and Discussion

5.1 Hole quality

In this study, carbide 0.5mm drill bits, stainless steel SUS304 and hardened tool steel SCM4 with 1mm thickness are used as specimens.

Fig. 7 and Fig. 8 show cutting conditions, work materials, original images of holes according to the drilling step. Diameter deviation Δd_e of each specimen are listed in Tables 2~3. These results show that location error and diameter deviation increase slightly as drill wear propagates.

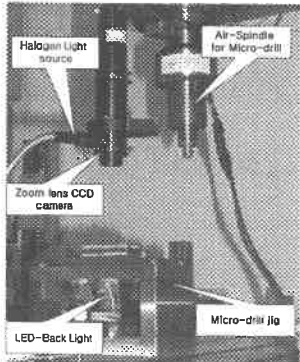


Fig. 4 Photo of experimental setup.

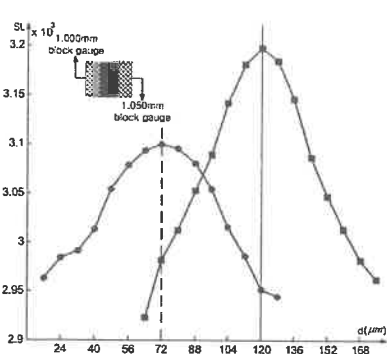


Fig. 5 SL focus measure function.

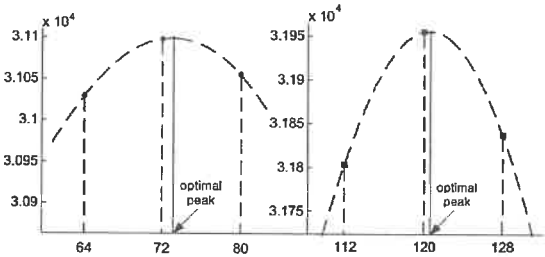


Fig. 6 Gaussian interpolation of focus measures.

Table 1 Calibration results of the developed SFF.

	Error
SL Focus Measure	$2.1 \mu\text{m}$
Gaussian Interpolation Focus Measure	$1.7 \mu\text{m}$

5.2 Burr width and height

To measure the exit burr heights, Δd of $8 \mu\text{m}$ and window size of 3×3 were empirically selected. Fig. 9 and 10 show original images of burr, measuring points and edges of specimens through image processing. Inspection results of burr geometry are listed in Table 4~5. “-” sign means the burr is located below the reference plane.

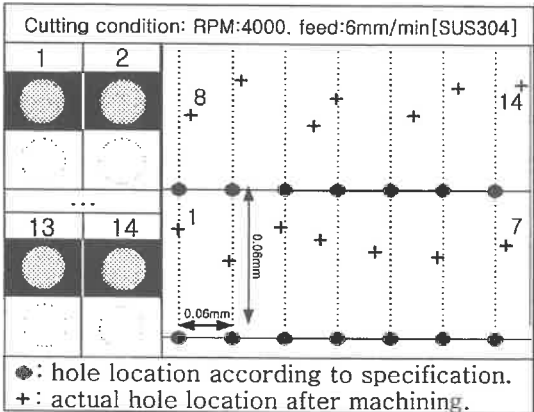
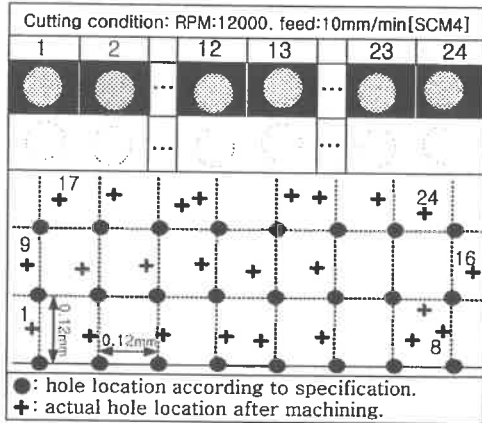


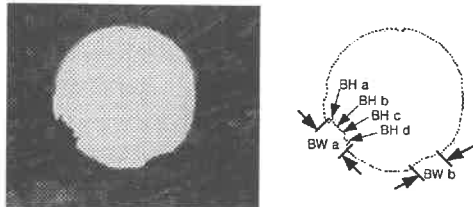
Fig. 7 Drilled hole images and location error map.

Table 2 Hole oversize result of SUS304.

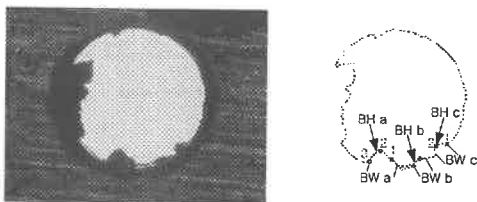
Hole Oversize (Δd_e): [μm]									
No	Δd	No	Δd	No	Δd	No	Δd	No	Δd
1	15.5	2	14.4	3	14.4	4	11.4	5	12.2
6	13.2	7	11.6	8	9.3	9	9.0	10	10.7
11	12.6	12	11.4	13	13.4	14	13.6		

**Fig. 8** Drilled hole images and location error map.**Table3** Hole oversize result of SCM4.

Hole Oversize (Δd_e) [μm]									
No	Δd	No	Δd	No	Δd	No	Δd	No	Δd
1	11.4	2	12.8	3	11.7	4	13.2	5	10.8
6	9.3	7	10.6	8	10.1	9	10.1	10	11.1
11	12.2	12	12.8	13	12.6	14	13.5	15	13.2
16	13.4	17	13.7	18	13.4	19	13.2	20	14.3
21	13.4	22	14.2	23	13.9	24	17.2		

**Fig. 9** Burr height(BH) and width(BW) of SUS304.**Table 4** Burr height(BH) and width(BW) [μm].

Burr height (BH), [SUS304]				
Reference	a	b	c	d
0	192.5	196.5	192.8	230.2
Burr width (BW)	a		b	
	117.7		107.6	

**Fig. 10** Burr height(BH) and width(BW) of SCM4.**Table 5** Burr height(BH) and width (BW) [μm].

Burr height(BH) [SCM4]						
Reference	Burr a			Burr b	Burr c	
	1	2	3		1	2
0	71.5	80.6	30.7	-80.6	-68.5	-80.2
Burr width (BW)	Burr a			Burr b	Burr c	
	108.5				67.9	60.7

6. Conclusions

In order to develop a condition monitoring system for micro-drilling processes, on-machine machine vision system is proposed to measure the burr geometry and the hole quality produced in micro-drilling processes. The machine vision system consists of a CCD camera with zoom lens, a novel illumination unit and image processing units.

Edge detection algorithms are used to measure hole quality. The SFF method is used to measure burr heights and widths. Developed system has resolution of 0.1 μm and accuracy of less than 2.1 μm in the measurement of burr height and width. The reliability of the developed system is confirmed through experiments.

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Fabrication of the Double Shell 2 Laser Targets Via the Bonding of Hemispheres

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Introduction

This paper describes the manufacturing method used to fabricate a meso-scale assembly that will be used as part of an experiment in high energy density physics. The assembly, known as the Double Shell 2 Laser Target, consists of two hollow, concentric spherical shells that are separated by a layer of carbon resorcinol formaldehyde aerogel with a density of 0.1 g/cm^3 . The inner shell has a diameter of 592 microns and thickness of 38 microns, and an aerogel structure holds the inner shell in place inside the outer shell, which has a diameter of 1254 microns and a thickness of 125 microns and consists of two hemispherical shells (hemis) that are bonded together at a step joint. A schematic diagram of the target design appears in Figure 1. There has recently been a great deal of interest in the development of cutting tools and processes for micro-scale and meso-scale mechanical machining [1-6], but there has been relatively little documentation on the practicalities encountered in transforming these laboratory capabilities into production efforts to machine and assemble meso-scale engineered parts. This paper presents the details of such a manufacturing effort.

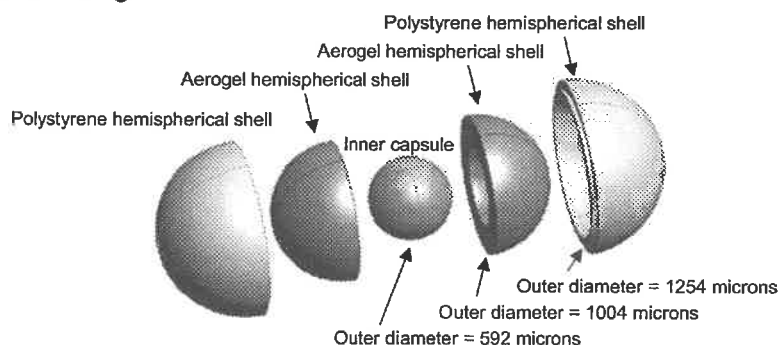


Figure 1. Exploded view of the double shell target

The completed double shell target will be shot with a high power laser, causing it to implode and compress the material inside the inner capsule. The reaction of the compressed material will be diagnosed, and the data will be used to validate physics computer codes. As with all laser experiment targets, this double shell target has strict requirements on its geometry and the precision with which it is fabricated. The target must be spherically symmetric, meaning the thickness of each shell should be uniform, the two shells should be concentric, and each shell should be smooth and free of defects. Tolerances on the absolute dimensions, sphericity, concentricity, and wall thickness uniformity are only a few microns.

Several previous researchers have successfully fabricated meso-scale double shell targets for high energy density physics experiments, and a variety of different manufacturing processes have been reported in the literature [7]. One method of producing outer hemispherical shells is to first create a hemispherical mandrel with an outer diameter equal to the desired inner diameter of the hemi [8,9]. The mandrel is then coated with the appropriate material, and the outer contour of the hemi is machined on its surface. The mandrel is then leached away, leaving a freestanding hemi, but the hemi can experience a dimensional change when the mandrel is removed. In another method, the outer shell is made in a single piece by coating a paraffin ball and subsequently removing the paraffin through a capillary [10]. However, the hole used to remove the wax must be plugged and creates a defect in the spherical shell. The inner capsule has been suspended inside the outer capsule using wires [10], plastic films [8,9], and foams that are backfilled prior to machining [9]. Each of these methods caused either non-uniformities in the targets or concentricity errors.

The manufacturing plan for the double shell target described in this paper was designed to produce targets that match the characteristics desired by the physicists as closely as the available resources would allow. In addition, the manufacturing process is deterministic and repeatable, and it does not suffer some of the dimensional instabilities and nonuniformities experienced in previous efforts. The following sections describe the construction of the components that make up the double shell target from the center toward the outside.

Inner Capsule

The inner capsule was manufactured by an external vendor. No additional machining was performed on this capsule.

Aerogel Hemispherical Shells

Two aerogel hemis support the inner capsule inside the outer shell. The space between the two capsules must contain as little mass as possible and be spherically symmetric. To provide a support structure of the smallest possible density, the inner capsule was held in place using aerogel [11]. To maintain spherical symmetry, the support was made in the form of two identical hemispheres with an inner contour to fit the inner capsule and an outer diameter equal to the inner diameter of the outer shell.

The hemis were machined on a Precitech Nanoform 200 diamond turning machine equipped with an air bearing spindle and a B-axis that allows the tool to rotate about its nose. All machining was performed with ambient temperature control of better than 0.5 degrees Celsius. The machine tool has a specified positional resolution of a few nanometers, and the absolute positional accuracy over the working volume of interest is estimated to be better than 0.2 microns. The workpieces were mounted in special holders that allow parts to be removed and reinstalled in the machine with a positional repeatability of approximately 0.3 microns. The tools were single crystal diamonds with a nose radius of 5.2 microns. To avoid crumbling or gouging the aerogel material, the feed was limited to 0.19 microns per revolution with a depth of cut of 2.5 microns and a spindle speed of 1600 rpm.

Each hemi was fabricated by first gluing a piece of the aerogel to a plastic arbor. The aerogel was rough machined to a cylindrical shape before machining a partial sphere onto the end, as illustrated in Figure 2a. The partial sphere was then broken off using a surgical blade and carefully inserted into a specially prepared vacuum chuck (Figure 2b). The vacuum chuck was then placed in the diamond turning machine, and a workpiece coordinate system was established by touching the tool to the face of the vacuum chuck. The portion of the sphere extruding from the vacuum chuck was faced down until only a hemisphere of aerogel remained, and then the inner hemispherical cavity was machined (Figure 2c).

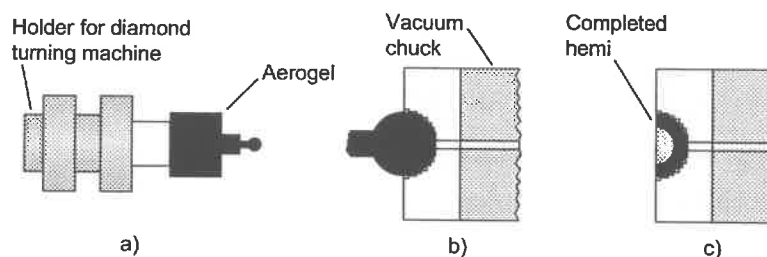


Figure 2. Machining process for the aerogel hemis

Outer Ablator Hemispherical Shells

The manufacturing process for the outer ablator hemis is illustrated in Figure 3. To fabricate each hemi, a piece of polystyrene was glued to a holder, placed in the diamond turning machine, and machined to a concentric cylinder (Figure 3a). Next, the internal hemispherical cavity and the joint were machined using the same diamond tool used to machine the aerogel (Figure 3b). For a roughing pass, the feed per revolution was generally 0.19 microns with a depth of cut of 2.5 microns. For a finishing pass, the feed per revolution was generally 0.038 microns with a depth of cut of 1.2 microns.

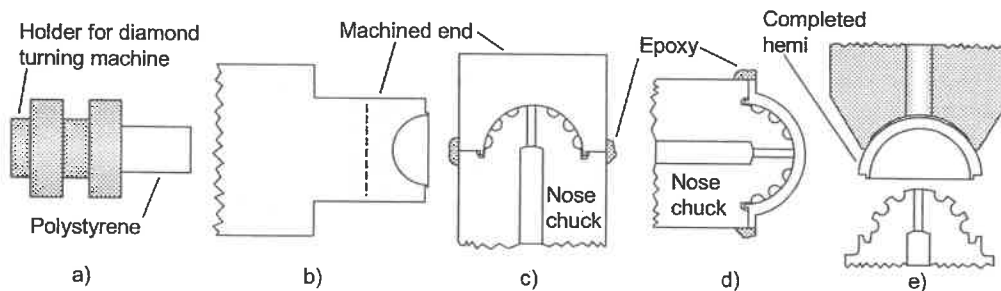


Figure 3. Machining process for the polystyrene hemis

After machining the internal contour and the joint, the machined end of the cylinder was cut off using a parting tool. This machined end was then placed over a nose chuck and held with vacuum (Figure 3c). The nose chuck consisted of a series of vacuum channels that were precisely machined to match the internal contour of the hemi. To augment the holding force of the vacuum during the roughing cuts, the machined cap was glued to the nose chuck by applying epoxy around the circumference of the interface. After the epoxy had cured, the part was placed back into the diamond turning machine, and a workpiece coordinate system was established by touching the tool to a pre-machined datum on the nose chuck. The external contour of the hemi was then machined in two steps using a tool with a nose radius of 25 microns. In the first step, the epoxy bond was left intact to provide strength while machining the bulk of the hemisphere. For this operation, the feed was 1.6 microns per revolution, and the depth of cut was 2.5 microns. This step left only a small flange of material at the interface of the part and the nose chuck (Figure 3d). In the second step, the epoxy and flange were machined away using a lighter depth of cut of 1.3 microns, leaving only the vacuum of the nose chuck to hold the part in place. This final machining step also required that the tool cut into the nose chuck, so a separate chuck had to be fabricated for each outer hemi. After completing the machining process, the finished hemi was removed from the nose chuck using an assembly vacuum chuck (Figure 3e).

Assembly Process

The components were assembled using a specially designed manipulator that uses manual micrometer screws to position the parts with micron level accuracy. Two perpendicular microscopes were used to align the parts radially before bringing them together axially. To prevent crushing the parts during assembly, the manipulator utilizes an air bearing slide coupled with a spring to provide axial compliance, and a strain gage transducer provides force feedback. The force transducer allows the operator to determine when mating components have made contact, and the assembly forces can be controlled to within a few milliNewtons.

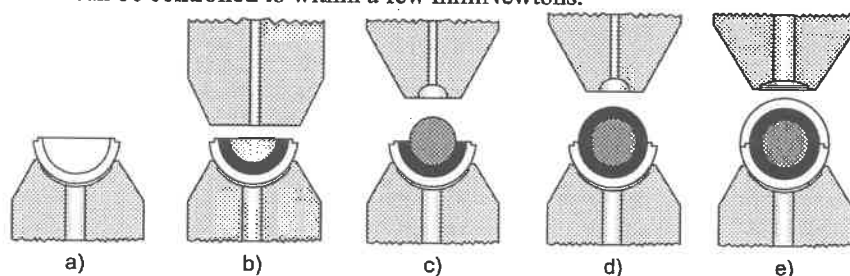


Figure 4. Assembly process

The assembly steps for the target are illustrated in Figure 4. First, the lower outer hemi was placed into the manipulator (Figure 4a). Next, one of the aerogel hemis was inserted (Figure 4b), and then the inner capsule was placed into the contour in the aerogel (Figure 4c). The other aerogel hemi was then placed on top of the inner capsule (Figure 4d). Finally, the upper outer hemi was placed on top to complete the assembly (Figure 4e). The assembly vacuum chuck was used to press the outer hemis together with a force of 150 milliNewtons while they were bonded. To join the two outer hemis, the joint was glued by applying three spots of cyanoacrylate adhesive in 120 degree intervals by hand using a single camel hair on the end of a stick. A droplet of the adhesive on the camel hair has a diameter of 50 to 100 microns, so the volume of the droplet is between 0.5 and 4 nanoliters. The gap in the joint has a volume of 0.5 nanoliters, so excess adhesive was usually applied. The adhesive wicked into the 2 micron gap of the joint and cured to form a secure bond. The excess adhesive spread to a thickness that was too small to be measured with the available equipment, and it was simply left on the surface of the target.

Characterization of the Assembled Target

A picture of the completed target appears in Figure 5, along with two radiographs of the completed target. One radiograph was taken normal to the joint, and the other was taken in the plane of the joint. The radiographs reveal a radial concentricity error of 2-3 microns, based on centers calculated from the outer diameter of the inner capsule and the inner diameter of the outer shell. This misalignment lies within the specified tolerance of 5 microns, and it correlates to a concentricity error of $\Delta r/r = 0.3\%$. Therefore, the current method of supporting the inner capsule with machined hemis of aerogel appears to be an adequate means of obtaining concentricity.

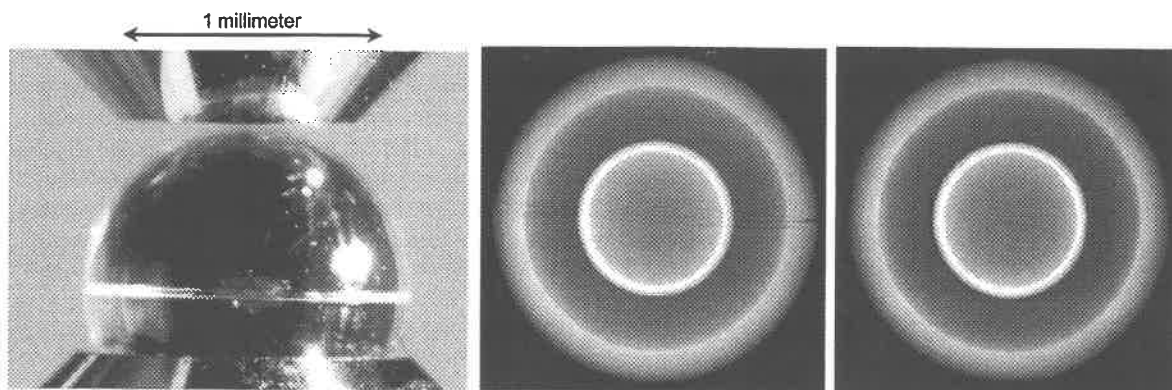


Figure 5. Completed target in a vacuum chuck (left) and radiographs of the target taken normal to the joint (center) and parallel to the joint (right)

Conclusions

This paper discusses techniques used to manufacture a precision meso-scale assembly consisting of machined aerogel and polymer hemispherical shells. Meso-scale manufacturing issues have been identified relating to micro-mechanical machining, part handling, fixturing, assembly, and bonding. Aerogel can be deterministically machined using a depth of cut on the order of a micron to form nearly massless meso-scale structures. Vacuum chucks provide an excellent means of fixturing meso-scale components when machining both internal and external contours. In cases in which the tool exerts a large force or torque on the part, the holding force of the vacuum can be augmented by gluing the workpiece to a sacrificial vacuum chuck. The assembly of meso-scale components is complicated by part alignment and the fragility of the parts. The bonding of the components was one of the crucial steps in the manufacturing plan, and this effort would have benefited from an ability to control the amount of adhesive dispensed with picoliter resolution and place the adhesive on the part with micron-level accuracy.

Acknowledgement

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Precision Manufacturing of Inertial Confinement Fusion Double Shell Laser Targets

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1. INTRODUCTION

Lawrence Livermore National Laboratory (LLNL) manufactures laser targets for experiments on the Omega laser at the University of Rochester and is preparing to build targets for the National Ignition Facility (NIF). These targets serve university collaborations, high energy density physics studies, and inertial confinement fusion (ICF) customers. Of particular interest to ICF studies are high-precision double shell implosion targets for demonstrating thermonuclear ignition without the need for cryogenic preparation. The suite of double shell ignition designs for the NIF [1] consists of a low-Z outer shell that absorbs laser generated x-rays, implodes, and then collides with a smaller high-Z inner shell containing the high-pressure deuterium-tritium fuel. Because the ignition tolerance to interface instabilities is rather low, the manufacturing requirements for smooth surface finishes and shell concentricity are particularly strict.

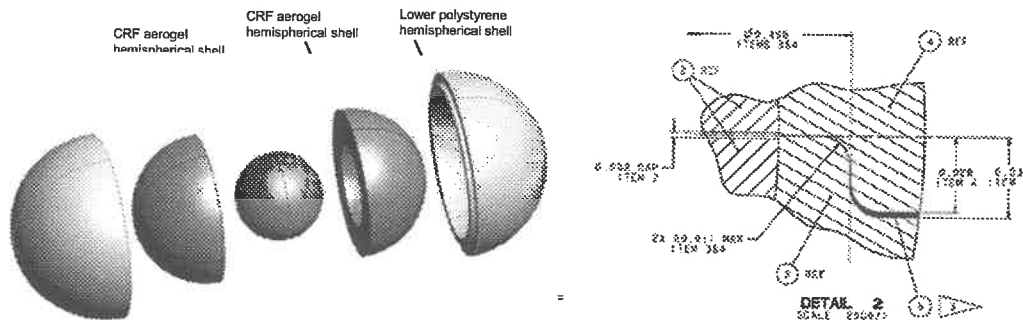


Figure 1a and 1b. Exploded view of the double shell target, a joint detail.

An exploded view of the double shell target appears in Figure 1a. The design consists of an inner plastic capsule with an outer diameter (OD) of 244 μm and a wall thickness of 15 μm . The inner capsule is suspended in two hemispherical shells (hemis) of low density CRF aerogel with a thickness of 220 μm . The outer ablator hemis have an outer diameter of 550 μm and are 52 μm thick. They are made of an LLNL fabricated 1 at % bromine doped polystyrene and mate at a step joint. A bonded joint secures the two ablator hemis together, Figure 1b.

The following capsule parameters are particularly important to these implosion experiments: concentricity of the two shells, thickness uniformity of the inner and outer shells, roughness of the various spherical surfaces, void volume fraction of the cyanoacrylate in the outer shell bond joint. The overall requirements are to maintain the dimensional tolerance of each component to $\pm 1 \mu\text{m}$ and all internal flaws to less than $\pm 0.5 \mu\text{m}$. The specification on shell concentricity is important for constraining the degree of performance degradation from an asymmetric shell collision. Similarly, any flaws or non-uniformities in the outer shell or bond joint can cause deviations from a spherical implosion, leading to performance degradation.

The objectives of this paper are to show how LLNL's precision engineering team approached the manufacture of a double shell target with a deterministic manufacturing plan and to discuss process limitations. An error budget was used to quantifiably study the manufacturing process and identify the effects of errors in each step. The manufacturing steps will highlight how each component is built with known datums and reference surfaces, and how the manufacturing plan created a deterministic, repeatable process. By taking this rigorous approach to controlling all error sources during machining and assembly, one can attempt to achieve the requirements of $\pm 1 \mu\text{m}$ dimensional accuracy, sub-micron surface flaws, and 5 μm concentricity between the two shells. This contrasts with previous literature in the 1980's and the 1990's that described the manufacture of double shell targets using methods that introduce uncertainties into the manufacturing process, such as coating mandrels and releasing free-standing components, or backfilling materials to allow machining of components, then extracting the filler material and dealing with part shrinkage [2, 3].

2. MACHINING OF THE COMPONENTS

To fabricate each target, the CRF hemis are machined, the inner contours of the ablator hemis and the step joint are machined, the five components are assembled and bonded, and then the outer contour of the target is machined. The following discussion of the machining processes will follow this sequence. This sequence of manufacturing steps differs from recent efforts at LLNL to manufacture larger double shell targets, in which

freestanding ablator hemis were placed around the CRF hemis and inner capsule and then bonded together [5]. This new manufacturing method has many advantages, including the ease of the bonding process, less precision is required in that an excessive amount of adhesive can be applied since it is machined off anyway, it improves the fit and finish of the outer contours since it is machined after it is bonded, and it improves the precision of design by not having to transfer and handle individual free standing hemis and position them in a vacuum chuck. The design philosophy behind this manufacturing plan is to know the reference position and size of each component throughout the manufacturing process and to minimize points where dimensional uncertainty can affect the quality of the target.

2.1 Vacuum chucks

Vacuum chucks are used in three places during this target build. The manufacturing method is similar for each of the three chucks and is illustrated in Figure 2. To create the vacuum chuck that holds the CRF, a hole is

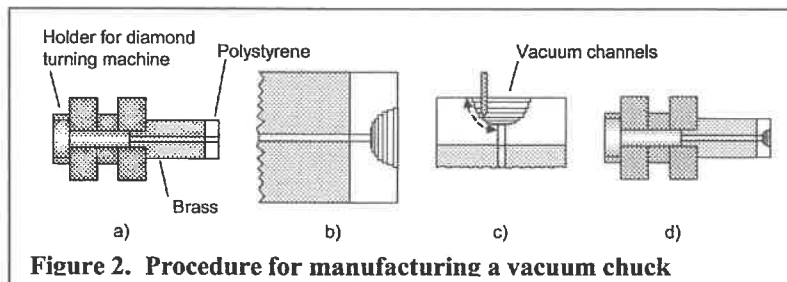


Figure 2. Procedure for manufacturing a vacuum chuck

drilled in a brass rod with a polystyrene disk on the end, which is mounted on a precision holder that allows vacuum to be pulled through the spindle of the machine, (Figure 2a). Then a hemispherical contour with a diameter slightly smaller than the OD of the CRF hemis is machined into the polystyrene disk, and a set of concentric channels 97 μm wide and 15 μm deep is cut into

the inner contour (Figure 2b). Then a groove is scratched into the contour to connect each of the channels (Figure 2c). The part is subsequently placed back in the DTM, and the contour is machined to its final diameter of 446 μm to match the OD of the CRF hemis (Figure 2d).

2.2 CRF hemispheres

Figure 3 illustrates the process for machining each CRF hemi. First, a large piece of the CRF is glued to an arbor mounted to a precision holding fixture. The CRF is then turned down to create a partial sphere of diameter 446 μm , supported by a neck of material, which will be the outer contour of the hemi (Figure 3a). The sphere is manually broken off at the neck using a surgical blade, and placed into the vacuum chuck (Figure 3b). Throughout the machining of the CRF, its position is deterministically known with respect to the DTM by orienting the hemis using vacuum chucks with reference surfaces. The only occasion in which the orientation of the part is not controlled with respect to a reference datum is when it is transferred to the vacuum chuck. To relocate the part once it is in the vacuum chuck, the tool is touched off on the face of the vacuum chuck, which is a known reference. This touch-off process allows the position of the part to be reestablished. The sphere is then faced off to create a solid hemisphere, and the internal contour is machined into the CRF to complete the hemi (Figure 3c). Each internal contour is custom fit to a particular capsule size. The length of the excess neck material is minimized to reduce the torsional moment on the vacuum chuck imposed by the tool cutting forces. When complete, the parts are inspected under a microscope to measure the ID. The uncertainty in this measurement is $\pm 1 \mu\text{m}$.

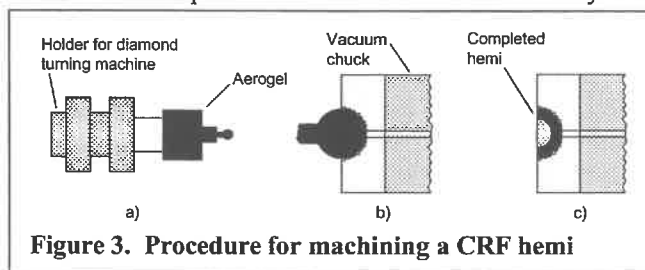


Figure 3. Procedure for machining a CRF hemi

2.3 Polystyrene ablator components

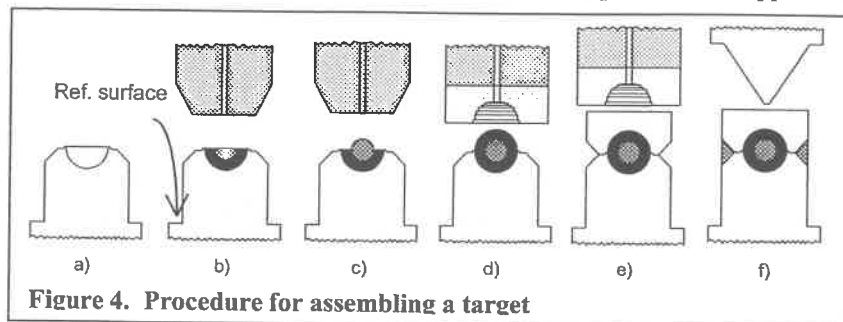
The ablator hemis are machined in two different stages. First, the inner contour, a reference surface and corresponding step joint are machined, as shown in 4a. The second stage is the machining of the outer contour of the ablators, but this is performed after the target has been assembled and bonded and is discussed in Section 4. This sequence for building the targets has simplified the

manufacturing and improved the quality of the target, compared to LLNL's previous double shell target design [5].

3. ASSEMBLY AND BONDING

An assembly fixture that uses the same workpiece holders that are used on the diamond turning machines is used for assembly. The assembly fixture is on a clean de-ionized bench and provides force control to 1 gram resolution. The assembly process is illustrated in Figure 4. Initially, the lower ablator component is inserted into the lower actuator of the assembly station (Figure 4a). Then one of the CRF hemis is placed into the lower ablator component (Figure 4b). Because the equator of the hemi is flush with the surface of the vacuum chuck, the part can be located axially by bottoming out the face of the vacuum chuck on the inner step of the joint on the lower ablator component. The inner capsule is then inserted into the inner contour of the CRF hemi (Figure 4c). The inner

capsules of the target are composed of plastic and are supplied by General Atomics (GA). The designed diameter of the capsules is $240 \pm 0.5 \mu\text{m}$ and varies from 239 to $247 \mu\text{m}$. The capsules are measured by GA by taking three equatorial traces around the sphere using a spheremapper [4]. The upper CRF hemi is then taken directly from the



vacuum chuck in which it was machined and placed on top of the inner capsule (Figure 4d). The inner contours of each CRF hemi are machined to match the outer diameter of the corresponding capsule, but due to the uncertainty in the OD of the capsule, the potential exists for concentricity errors at this step. In the final assembly step, the upper ablator

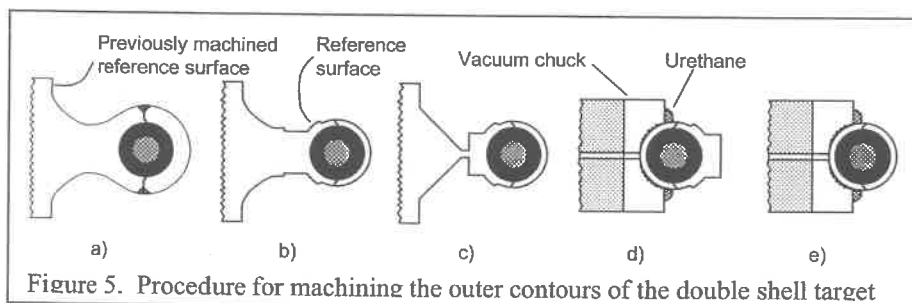
component is placed over the CRF (Figure 4e) and mated with the joint in the lower component. The two ablator components are pressed together with a force of 10 grams, and the seating of the joint is visually inspected to ensure it has a uniform gap around the perimeter.

The bonding of the ablator components is done on two steps. First, cyanoacrylate is applied around the perimeter of the joint by placing a droplet on the end of a camel hair and running it around the perimeter. The volume of cyanoacrylate applied is sufficient to fill the $2 \mu\text{m}$ gap and wick into the joint, but it fills only the apex of the tapered sections. The joint has a straight step, and the inside surfaces of the joint are flush to prevent the adhesive from wicking into the CRF. Then the tapered sections of the components are filled with epoxy (Figure 4f), which is allowed to cure for 24 hours. This epoxy supports the joint during machining and is fully machined away in subsequent steps. By machining the external surface after bonding the components, any crazing that occurs upon application of the cyanoacrylate is removed in subsequent machining, leaving a diamond turned quality surface.

4. FINAL MACHINING

There are three steps to the final machining: setting the tool, machining the first half of the outer contour and then transferring the target to a vacuum chuck, and machining the second half of the outer contour. The tool setting is critical for this operation, because any error in its position will cause wall thickness errors in the target. Therefore, before machining the outer contours of the ablators, the tool is reset.

Figure 5 illustrates the procedure for machining the outer contour of the target. The tool is touched to the previously machined reference surface on the lower ablator component to establish a workpiece coordinate system. Then the bonded assembly is machined from the form shown in Figure 4f to the form shown in Figure 5a. In this machining process, the tool also machines the OD of the shank near the reference surface. This machined OD



serves as a reference diameter that can be measured with an uncertainty of $\pm 1 \mu\text{m}$. Thus, by measuring this reference diameter, the OD at the equator of the machined target can be inferred. Note that at this stage, the support shank diameter and length have been maintained

to provide support, so the part will not deflect under the cutting forces. The epoxy inside the taper provides support to the bonded joint during machining. The free end is then machined to its final spherical form, and it includes another reference surface near the stem, as illustrated in Figure 5b. The stem holding the partially machined sphere is then reduced in size to a thin neck (Figure 5c), which is then broken off with a surgical blade. The free spherical end of the target is then gripped by the specially crafted vacuum chuck (Figure 5d), which holds the part while the remainder of the outer ablator surface is machined. To augment the grip of the vacuum chuck, a bead of urethane adhesive is applied to bond the part to the chuck (Figure 5d). The remainder of the outer surface of the target is then machined (Figure 5e), and the final finishing pass blends together the surfaces on the outer contour.

5. CHARACTERIZATION

A completed target is shown in Figure 6. Once the target is completed, only external dimensional metrology and characterization can be performed. Visual inspection of the joint using an optical microscope

indicates that the cyanoacrylate has fully wicked around the joint, and there are no visible voids or bubbles in the 2 μm gap. The seating of the two hemispheres appears uniform on all targets. The area in which the two outer ablator surfaces were blended together is inspected with a WYKO NT8000 scanning white light interferometer. It is to quantify the characteristics of the blend line, but the blended area appears to contain a surface flaw in the form of a step estimated at less than 0.3 μm on all targets (radial perturbation < 0.1%).

One of the most important characteristics of the targets is the concentricity between the ablator shells and the inner capsules, which is

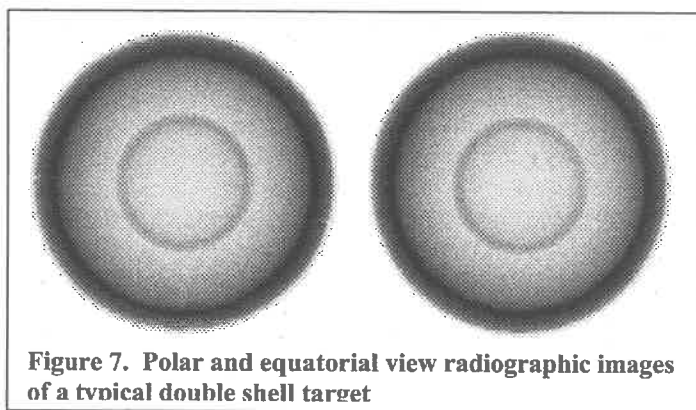


Figure 7. Polar and equatorial view radiographic images of a typical double shell target

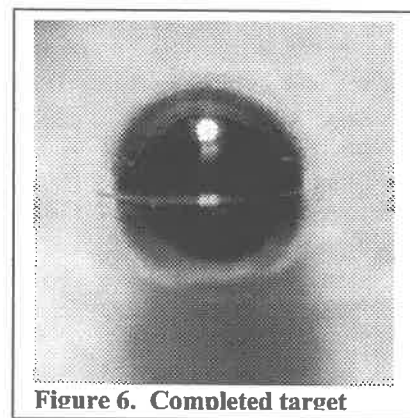


Figure 6. Completed target

determined from contact radiographs of the completed targets, (estimated resolution of approximately 1 μm .) Two orthogonal radiographs are taken of each target. Example radiographs for one of the targets appear in Figure . For each target, two different fitting routines are used to calculate the concentricity errors in the plane of the joint and in the pole-to-pole direction. The six targets have concentricity errors of 1, 3, 4, 4, 5, and 6 μm , which represent a $\Delta r/r$ of between 0.4% and

2%. Only one of the six targets does not meet the specified concentricity of 5 μm .

6. SUMMARY

By taking a rigorous approach to controlling the manufacturing process, double shell targets have been manufactured to meet the specifications of ± 1 μm dimensional accuracy and 5 μm concentricity. These targets provide physicists with an experimental target that may yield improved performance. This approach of using known manufacturing methods and pushing the limits of conventional processes can achieve high quality parts, because the manufacturing process has minimized the introduction of geometrical/dimensional uncertainties in each component. The reality is that process control and development are the major tasks in executing a job like this. Most of the effort involves paying extreme care to the minute details at each step.

Throughout this process, we have identified areas where improvements can take place, and the preliminary error budget has been refined to identify the most important sources of error and uncertainty in the manufacturing process and to reveal their impacts on the quality of the completed targets. The error budget reveals that the key areas are developing a better assembly station, reducing the uncertainty in the measured diameters of the inner capsules, obtaining better metrology tools, and obtaining a better tool set station.

ACKNOWLEDGEMENTS

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FINITE-ELEMENT SIMULATION OF THE UNIAXIAL DIE PRESSING OF CERAMIC POWDER

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I. Background and Motivation

Modern engineering ceramics such as alumina and zirconia are widely used in aerospace, automotive and electronics industries, but are often difficult and costly to process. We are currently developing novel fabrication techniques to manufacture a meso-scale ceramic heat exchanger [1,2,3]. This device made of Functionally Gradient Materials (FGMs) contains a complex internal network of channels to circulate cooling fluids. Two different approaches have been attempted. In the first approach, ceramic powder is poured into a steel mold, and internal meso-scale channels introduced into the powder bed by embedding a fugitive phase such as a graphite piece. The powder bed with the embedded fugitive phase is then uniaxially die-pressed to form the green compact, which is sintered to yield the final product. The fugitive phase burns away during sintering, leaving behind channels in the shape of fugitive phase in the bulk of ceramics. One particular concern in this approach is the limited powder flow around the fugitive phase. In the second approach, green compact without fugitive phase is partially sintered and subsequently machined to achieve complex surface channels.

The success of these fabrication techniques is dependent on the relative density of the compact achieved upon compaction and partial sintering especially meso-scale devices. Inhomogeneity in relative density distribution will result in sintering rate differences, which may lead to crack formation. In addition, density gradients in pressed-powder compact may result in severe shape distortion and unpredictable shrinkage that deleteriously affect the accuracy and reliability. Researchers like Aydin et al [4], Zipse [5], Bouvard et al [6], Kim et al [7,8] and Kim et al [9] have approached the compaction and sintering problems using the finite element method. In this paper, various specimens are manufactured and machined to aid in understanding the proper implementation of a finite element model. Ceramic powder bed with and without embedded fugitive phase, both subjected to uniaxial die pressing and ejection, are simulated using commercial software ABAQUS. The simulation results are then analyzed for a) the effects of orientation and locations of embedded phase on the relative density distribution and b) the plausibility of machining on a compacted without an embedded phase based on the simulated density distribution.

II. Specimens and Machining

In the current effort, the fugitive phase is the graphite piece EDMed to a cross-section of 2mmx2mm and then CNC-machined to produce controlled groove features of 0.8 mm wide and 0.67 mm deep as shown in Fig. 1(a). These groove features are repeated on all 4 sides along the graphite piece. The graphite piece is then embedded in the powder bed before compacting, shown in Fig. 1(b). Subsequent processing details such as die pressing and sintering can be found [1-3]. The sintered products, and the features imprinted, are shown in Fig. 1(c). The channel final dimensions have shrunk to 1.5mmx1.5mm. It is clear from Fig. 1(c) that the groove features successfully implant their negatives, or fins along the channels. It is our hope that these fins can

be manufactured in controlled dimensions and orientations so that meso-surface-texture engineering on the channels are possible for optimum heat transfer.

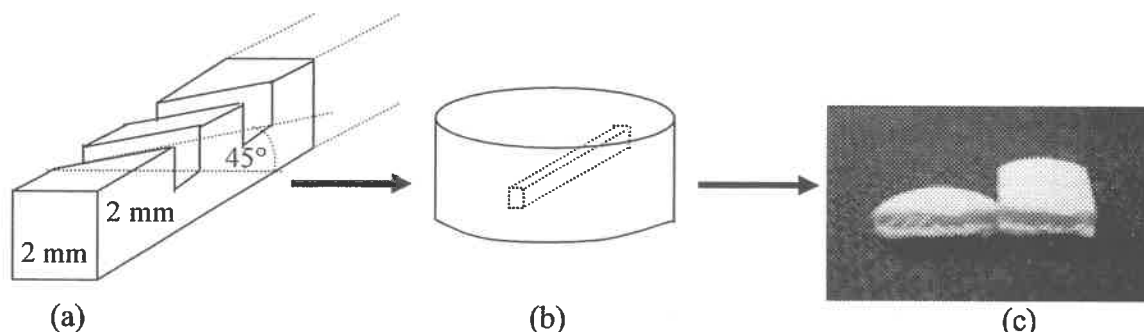


Figure 1: (a) EDMed and machined Graphite; (b) Graphite embedded in powder bed and compacted and (c) Final sintered product fractured to reveal the details.

III. Finite Element Simulations and Constitutive Model

A hyperbolic-cap elastoplastic model with a shear failure line (or critical state line) originally proposed by Kim et al. [7, 8] to model the zirconia compact behavior is employed in the compaction simulation. This model predicts hardening (as the cap yield curve expands, for example, from dash-dot to solid curve) under triaxial loadings by associative flow rule and non-dilational perfectly plastic deformation of the powder compact under shear loadings with a non-associative flow rule as shown in Fig. 2 where F_s is the shear failure line; F_c is the cap yield curve; p is the hydrostatic stress; q is the equivalent stress; d is the material cohesion, and β is the material friction angle. The stress state within the envelope assumes elastic deformation.

The model is implemented through a FORTRAN code UMAT in ABAQUS 6.3 with the algorithms proposed by Aravas [10] and Govindarajan [11]. The constitutive model, and the material properties of zirconia powder are detailed elsewhere [7-8]. This model is compared to simulation results using a similar model in ABAQUS internal library, namely the modified Drucker-Prager/cap model for verifications [4-6]. Due to the space constraint, the results from the hyperbolic-cap model only are presented as the simulated density distribution both models exhibit similar trend and magnitude, except for some local differences, which has to be further investigated through experiments. Based on literature resources [4-9], an arbitrary value of 0.17 for the Coulombic friction coefficient between the compact and powder wall in the 3D models, whereas in the axisymmetric and plain strain models a value of 0.2 are used. The graphite piece is assumed to behave elastically throughout the compaction process.

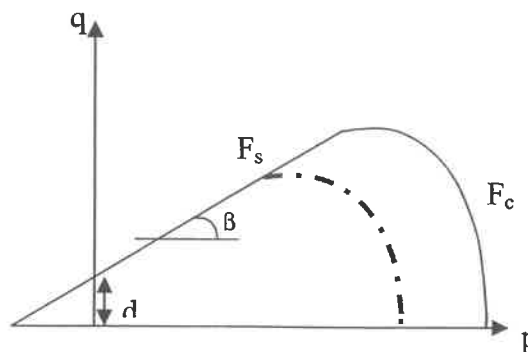


Figure 2: The hyperbolic cap model

IV. Simulation Results and Discussions

In particular, the following models were developed and simulated to understand the processing techniques described in Section II. These models are:

- Two axisymmetric models with different aspect ratios (h/D) as in Fig. 3(a) and Fig. 3(b).
- A 3D finite element model of the cylindrical powder compact with a rectangular graphite piece, as shown in Fig. 4; and
- A plain strain finite element model with graphite piece located in different position and orientations, as depicted in Fig. 5(a) (b) and (c).

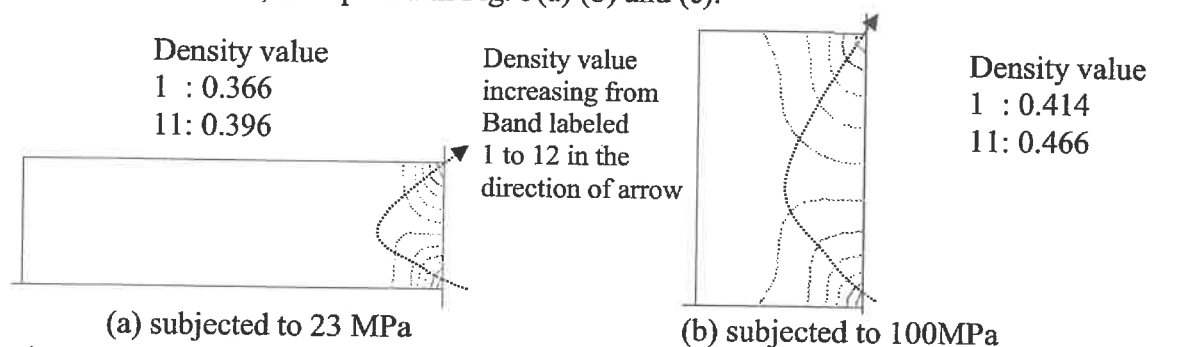


Figure 3. Relative density distribution after ejection for compact with different aspect ratio. (The contour bands in all results presented as follows are equally-spaced.)

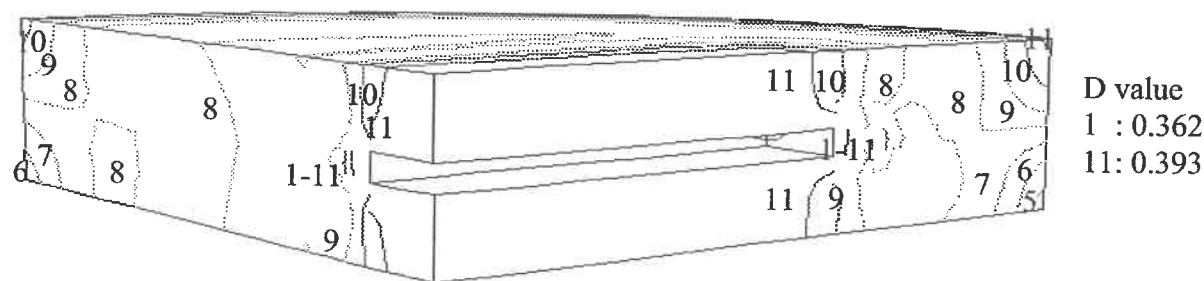


Figure 4: Relative density distribution after ejection for the 3D model, subjected to 23 MPa. The empty spot is where the graphite resides.

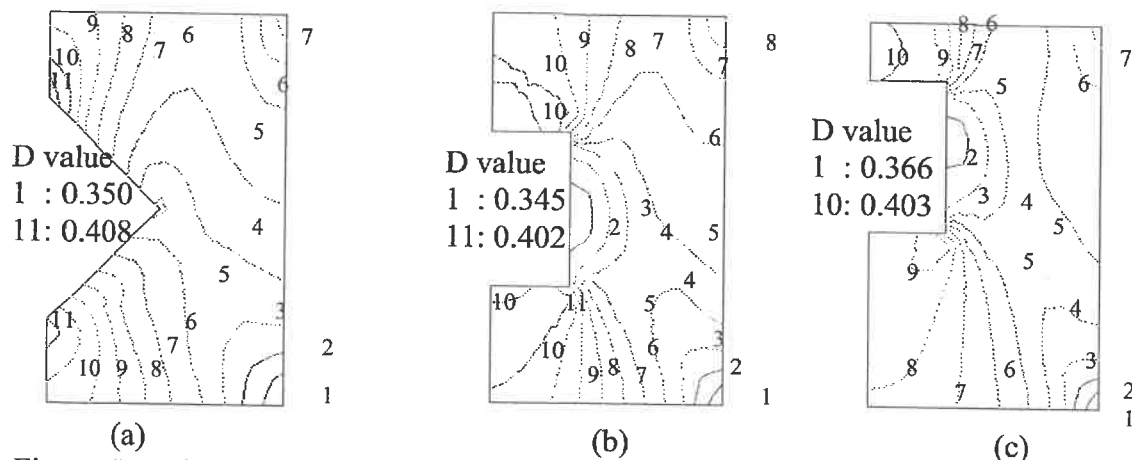


Figure 5: Relative density distribution after ejection with embedded graphite piece in different orientation and location. All compacts are subjected to 23MPa.

Inhomogeneity in density distribution is simulated in the presence of the graphite. Regions of high density gradients should be avoided to reduce possible defects on the final products. Figs. 3

show that a cylindrical compact with a low aspect ratio yields more uniform density distribution and thus allowing PSC machining on the most region of the sample. Fig 3(b) shows the regions of higher density at the top corners as compared to that at the bottom corners are very consistent with the findings of various researchers [4-9]. In Fig. 4, the region around the graphite piece has high-density gradients, and is thus a possible crack initiation location. Fig. 5(a), (b) and (c) seem to suggest that a diagonal graphite piece orientation and midsection location are preferable, since they tend to reduce density gradients.

V. Conclusions

The compaction simulation provides a valuable tool to understand the processes used in this study, mainly micro-machining and compaction with fugitive phase. Machining PSCs with a low aspect ratio was successful due to the uniformity in density distribution. Engineering the micro-surface-texture of meso-scale mixing channels was also introduced within a ceramic bulk by using a graphite fugitive phase. The geometry and positioning of the fugitive phase can be designed with the aid of FEM. The effects of powder bed dimension, fugitive phase dimension, its orientation and location on density distribution during powder compaction had been simulated. These results can help in designing the optimum geometry without introducing defects in the final sintered products.

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Precise gastight sealing for bio/chemical microchip

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1. Introduction

Recently, the use of micro-channels on microchips is catching much attention for miniaturizing and integrating bio/chemical reaction systems. The large surface area, per volume, gives high thermal conductivity to a micro-channel allowing quick and accurate temperature control of the reagents inside. In addition, its small size shortens the molecule diffusion time for fast and efficient reagent reaction. Furthermore, it requires smaller volume of reagents and reduces the waste fluid.

A micro-TAS (micro Total Analysis System) is a chemical analysis system that uses micro-channels. They are popular in immunoassay constrained by small sample sizes and they complete their tasks in very short analysis time[1]. In contrast, a system that performs chemical synthesis in a micro channel is a micro-reactor. A micro-reactor is highly efficient for thermally controlling reagents and is suitable for exothermic reaction, which requires cooling, and other reactions that need tight thermal control for putting a reagent into a specific temperature range for producing a target reagent.

Generally these microchips are manufactured by cutting grooves on substrates and covering them, however, a common problem is bonding a lid and a substrate of glass and metal with gastight sealing.

Glass is suitable material for micro-TAS's, so the researchers can visually observe the reactions[2][3]. Other transparent material, such as plastics, or silicone rubber, are also available, however, do not stand high temperature, and corrode when exposed to acid, alkali and organic solvents. In contrast, observing the reaction, perhaps, is not so important with chemical synthesis and the micro-reactor material does not need to be transparent. Therefore it should be made of metal that generally resists high temperature and pressure, and has excellent thermal conductivity. In addition, it is easy to form and is shock resistant. Some metal even resist acid, alkali, and organic solvent.

There are some method for bonding glass and

metal, diffused junction, thermal adhesion, chemical adhesion using hydrofluoric acid, however, they need high temperature, high pressure, and surface flatness in atomic order. In addition, since they are all whole surface junction, a lid could close the micro-channels covering the substrate.

To counter these problems, we propose precise bonding for not whole surface but only a local area around the micro-channels. The local bonding does not close the micro-channels and needs less heat and pressure. We employed laser welding for a metal microchip, and sintering a paste with glass flit for a glass microchip.

2. Laser welding for metal microchips

A metal microchip has to resist both high and low temperature, pressure, or stress. It also has to remain unaffected by such reagents as acid, alkali or organic solvent. Material that meet these requirement include, stainless steel, nickel, platinum, hastelloy, and inconel. We chose stainless steel for its ease of forming and low cost.

We then cut micro-channels on the surface of the stainless substrate using a fine end mill. The micro-channel size usually determines the method for machining them on stainless substrates. The channel size we needed was in the range of 200-1,000 micrometers in both width and depth. When larger, we can neglect the scale effect, however, for smaller channels, viscosity of the fluid causes flow resistance that cannot be neglected. Ferric chloride or aqua regia wet etching is also popular in cutting channels but only when the channels are 100 micrometers deep or shallower.

For the next process of covering the channels machined on the substrate, we laser welded a stainless steel plate. The specific machine we used was a Compact YAG Laser (ML-2030, MIYACHI Co., Ltd.), with a wavelength of 1064nm and a focus diameter of 500 micrometers. For finding the optimum laser output and welding conditions, we ran a preliminary test varying the laser voltage and irradiation time. Table 1 shows the results. For

welding a 0.1mm cover to a 5mm substrate, we found the optimum condition of laser output at 250-310V for 5-10msec (3.9-12.5J). Fig. 1 shows the cross-sectional view of the substrate welded with the laser output of 270V for 10msec (7.8J).

We produced a metal microchip with the processes mentioned above. Specifically, we cut 400 micrometers wide and 330 micrometers deep micro-channel, using a ball-end mill with radius 200 micrometers, on a 5mm thick SUS316L substrate. We then welded 0.1mm thick SUS316L plate on the substrate to cover the channels. A YAG laser with an output voltage of 270V and irradiation time of 10msec stopping every 400 micrometers performed the welding. Fig. 2 shows a photo of the metal microchip we produced.

We tested the welding quality by pressurizing the microchip to 220kPa, a pressure high enough pressure for general reaction, and holding the pressure for 20 minutes. A holder with inlet and outlet ports fixed the microchip in position. A fluoroplastic O-ring connected each port to a bolt with a pipe welded to it. Fig. 3 shows the pressure inside the microchip. No pressure drop was observed during this time demonstrating that the YAG laser successfully welded the microchip with gastight sealing.

Furthermore we tested the microchip by a temperature controlled reaction that mixed the reagents. The test demonstrated an exothermic reaction with acid, which is nitration of acetanilide. The local reaction temperature inside the microchip showed the good temperature controllability of the microchip.

Table 1 Results of the preliminary test varying the laser voltage and irradiation for welding a 100-micrometer cover to a 1mm substrate.

Irradiation time voltage	230[V]	250 [V]	270 [V]	290 [V]	310 [V]	330 [V]
2.0[ms]	△	△	△	△	△	△
5.0[ms]	△	○	○	○	○	○
10.0[ms]	△	○	○	○	○	×

○ : welded successfully
△ : not enough output for welding
× : make a hole on the top layer

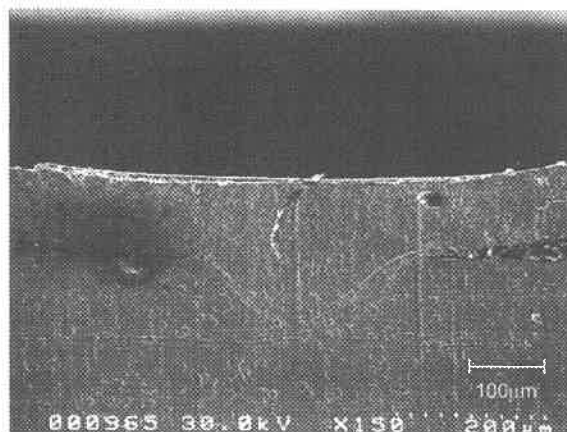


Figure 1 Cross-sectional view of the substrate welded with the laser output of 270V for 10msec (7.8J)

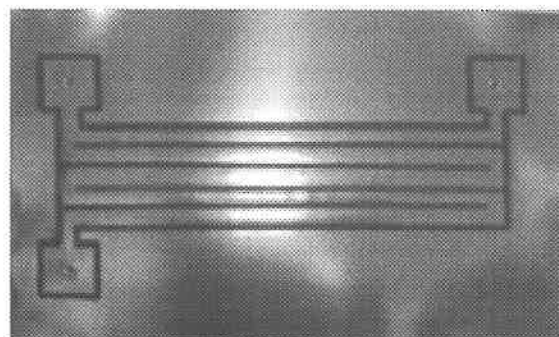


Figure 2 Photo of the laser welded metal microchip.

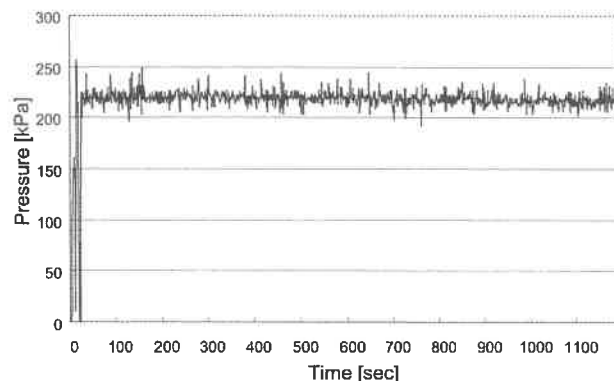


Figure 3 The pressure inside the metal microchip during pressurized to 220kPa and held for 20 minutes.

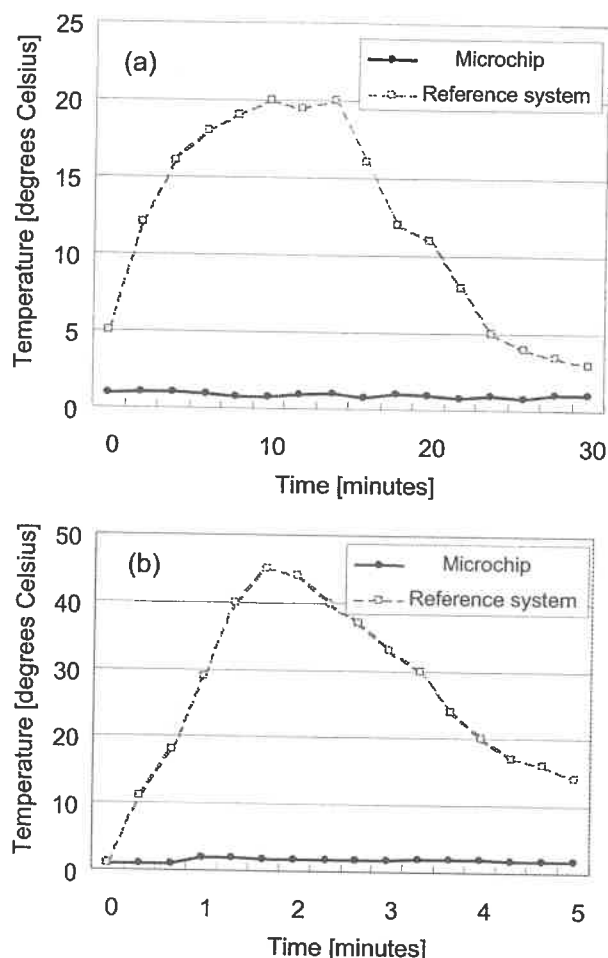


Figure 4 Temperature transition during (a) 30 minutes reaction, and (b) 5 minutes reaction.

The reaction proceeded as follows: First, 10 grams of acetanilide were dissolved in 30ml of sulfuric acid, and 12ml of nitric acid were added into the solution. For producing the target reagent p-nitroacetanilide, the microchip had to hold the reaction temperature at 15 degrees Celsius or less. Adding pure water to the mixture recrystallized the p-nitroacetanilide which was then separated by filtering. Syringe pumps fed the sulfuric acid solution of acetanilide, and the nitric acid. Thermocouples were attached to where the junction was for measuring the reaction temperature.

Fig. 4-(a) shows the temperature transition during a 30 minutes reaction of the two reagents, and Fig. 4-(b), that when we forced the reaction to complete in 5 minutes. We also took a reference measurement of the same reaction inside a much

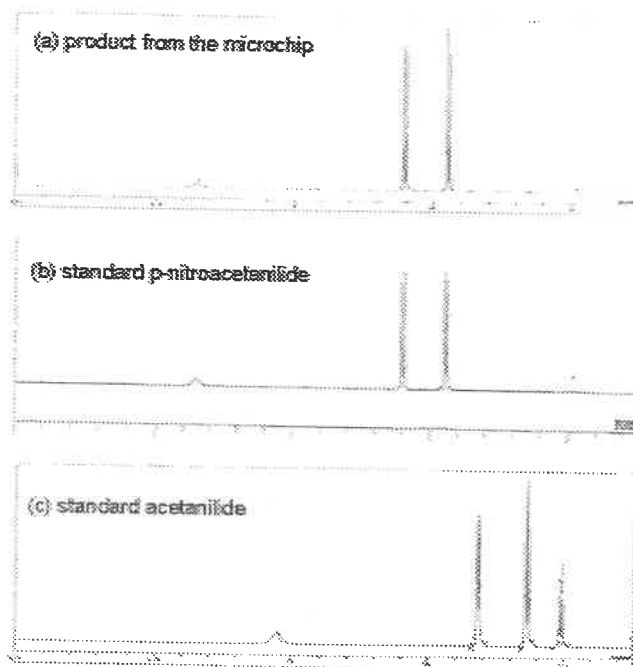


Figure 5 Analysis results of the composition of (a) the product from the microchip, (b) standard p-nitroacetanilide and (c) acetanilide using nuclear magnetic resonance.

larger beaker. Both graphs show that the reaction temperature values were kept at 3 degrees Celsius or less inside the microchip, whereas it went up to 20-45 degrees in the reference system.

We next analyzed the composition of the product from the microchip using nuclear magnetic resonance (NMR). Fig. 5-(a) shows the analysis results, and Fig. 5-(b) and Fig 5-(c) show the composition for standard p-nitroacetanilide and acetanilide. Comparing the peaks in the charts reveals that the microchip produced the required amount of p-nitroacetanilide without any residual. This test confirmed that the microchip succeeded in mixing and letting the reagents react while controlling its temperature.

3. Paste sintering for glass microchip

Direct glass junction needs high pressure and high temperature about 800-1400 degrees. Otherwise it needs dangerous process with hydrofluoric acid. Thus, we employed sintering process of a paste with glass flit which needs lower temperature, less pressure and no hydrofluoric acid. In addition, controlling the thickness of the paste layer can skip

the glass grooving process including complicated mask process.

Fig. 6 shows the bonding process of the glass microchip. We first put a channel-shaped polyimide film as a core on a 1 mm thick substrate. We then put a glass flit paste on the substrate along the channel core using a precise dispenser. Next, we put another 1 mm thick glass plate on the substrate and sintered them at 580 degrees for 1 hour. The core which is 500 micrometers wide and 150 micrometers thick decomposed during the sintering. The glass flit has a particle size of 50nm-2 μ m and a deformation temperature of 475 degrees.

Fig. 7 shows the sintered microchip. We measured the channel width and depth, and the bonding strength of the microchip. The result shows that the channel size was in the range of 460-540 micrometers in width and 140-160 micrometers in depth. It means that sintering process made a channel size error of 8% from the lost core. The bonding strength against tensile stress was 1.2 MPa, enough pressure for the general reaction. However, micro cracks, which causes leakage, were found on the sintered paste. They seemed to be produced when the solvent in the paste vaporized. Sintering the paste in vacuum for degasification will defeat this problem.

4. Conclusion

We developed two bonding method for metal and glass microchips. YAG laser successfully welded a SUS316L plate over the micro-channels to cover them. The welding quality was tested by pressurizing the microchip and holding the pressure. Our metal microchip performed well in nitration of an acetanilide. Paste sintering achieved glass bonding. We put glass flit paste along a polyimide core using precise dispenser. Sintered for 1 hour at 580 degrees, the glass microchip has a bonding strength of 1.2MP, however, it has a channel size error of 8%, and micro cracks on the sintered paste. The future work is to optimize the sintering conditions for bonding without cracks.

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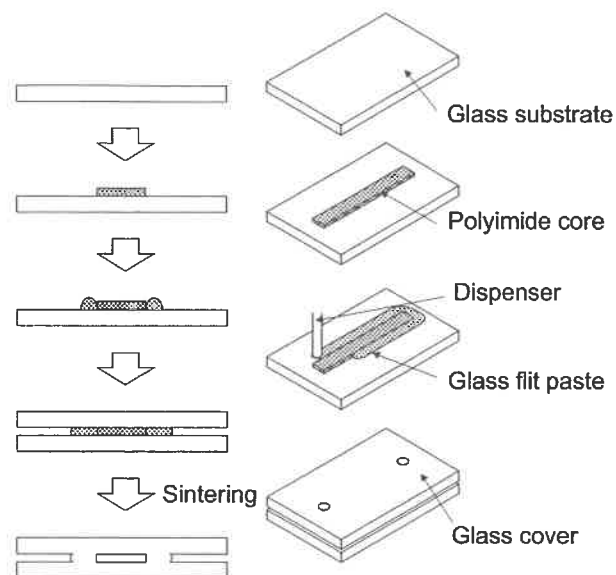


Figure 6 Configuration of the bonding process of the glass microchip by sintering glass flit paste.

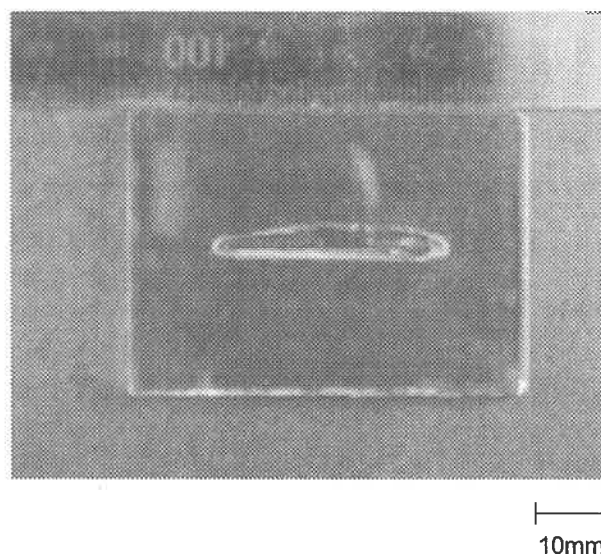


Figure 7 Glass microchip sintered for 1 hour at 580 degrees.

Algorithm to establish manufacturing process of new functional materials

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Abstract

The material with high applicable stability level is silicon and quartz as for the processing technology to which the material characteristic was investigated, and established for many years. The new crystal material artificially promoted is searched in world each place aiming at new functional materials. In general, these materials often have the limitation in an applicable manufacturing process compared with silicon and quartz. Lithium tetraborate ($\text{Li}_2\text{B}_4\text{O}_7$) is expected as a new functional material in an ultrasonic field and an optical field. Especially, the surface of the wafer is dissolved to the acid weakly to water as for $\text{Li}_2\text{B}_4\text{O}_7$ used as a surface acoustic wave (SAW) material. The pattern formation method by the lift-off was used. Establishing a steady manufacturing process from no use of water to the pattern formation had studied the difficulty thoroughly. This character that $\text{Li}_2\text{B}_4\text{O}_7$ is not violated by the alkali developer exists. The manufacturing process in $\text{Li}_2\text{B}_4\text{O}_7$ was able to be established by positively using this feature. The linear expansion coefficient has anisotropy in the side in $\text{Li}_2\text{B}_4\text{O}_7$. The improvement of the temperature characteristic was tried by changing it by positively using this character compared with the vicinity of the chip. The state that the size of the pattern on the surface of $\text{Li}_2\text{B}_4\text{O}_7$ where the pattern had been formed to the temperature change doesn't change was able to be produced. This new structured SAW device that $\text{Li}_2\text{B}_4\text{O}_7$ wafer is excellent from a basic performance to long-term stability and an environment that has is originally obtained.

Introduction

It is hoped to make to the device by using the material with extremely high stability level, and to suit the telecommunications system that assumes necessary in the ultrasonic electronics field. Development of the device that has the inquiry and the new feature of a new material for this Imperial command title in each place and a new system are examined. Here, it bases on to the material side, and it considers it. The technique of a solution and a further performance improvement of a technical problem were examined about the SAW. It is impossible near to obtain the one to satisfy all the performances at first the development of a new material almost. The problem of appearing newly can be solved very well becomes important. To realize the high stabilization, $\text{Li}_2\text{B}_4\text{O}_7$ was examined about SAW. The technique to have investigated this $\text{Li}_2\text{B}_4\text{O}_7$ can be applied as an algorithm to establish the manufacturing process to new functional materials.

Problems of $\text{Li}_2\text{B}_4\text{O}_7$

$\text{Li}_2\text{B}_4\text{O}_7$ is that the problem on the manufacturing process and the temperature change of the frequency are large as shown next.

1. $\text{Li}_2\text{B}_4\text{O}_7$ crystals are well known to dissolve in water and acid solution which is usually used in SAW device fabrication process. Figure 1 shows the dependence of the dissolution rate of $\text{Li}_2\text{B}_4\text{O}_7$ substrate on PH in a solution. The dissolution rate on $\text{Li}_2\text{B}_4\text{O}_7$ substrate is proportional to H^+ ion concentration in a solution.
2. The characteristic of the temperature of $\text{Li}_2\text{B}_4\text{O}_7$ SAW is examined and the method of obtaining the characteristic of the temperature at ST-cut quartz level is examined. (Refer to Figure 2.)

As for $\text{Li}_2\text{B}_4\text{O}_7$, the single polarization processing is unnecessary as shown in Table 1 because of the piezoelectrics. LBO, the width of the solid solution is narrow, and the composition change is few. The variation of 3-inch diameter $\text{Li}_2\text{B}_4\text{O}_7$ wafer at the SAW speed is fewer than that of LiTaO_3 and LiNbO_3 . The manufacturing variation of SAW chips in the wafer is small, the manufacturing yield is high.

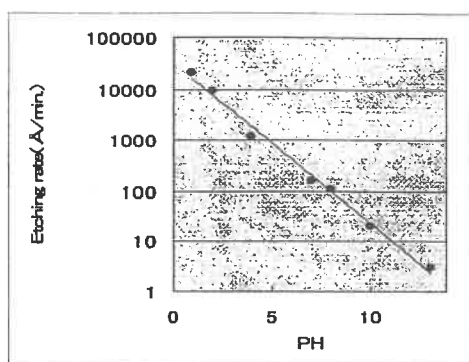


Figure 1 dependence of the dissolution rate of $\text{Li}_2\text{B}_4\text{O}_7$ wafers on PH

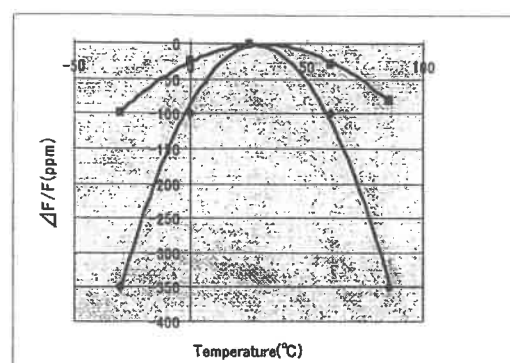


Figure 2 Frequency temperature characteristics of $\text{Li}_2\text{B}_4\text{O}_7$ SAW and ST-cut SAW

Table 1 Variation of SAW materials velocity

	Variation of SAW Velocity	Remarks
$\text{Li}_2\text{B}_4\text{O}_7$	<1m	Paraelectrics
LiTaO_3	2m~3m	Ferroelectric
LiNbO_3	5m~10m	Ferroelectric

Figure 3 shows progress recently at the time of compared the SAW material wafers with the silicon one. $\text{Li}_2\text{B}_4\text{O}_7$ has enlarged just like other materials for the cost reduction. $\text{Li}_2\text{B}_4\text{O}_7$ crystal has succeeded in the promotion of 8-inch diameter by the Brigeman method in the laboratory in 1992.

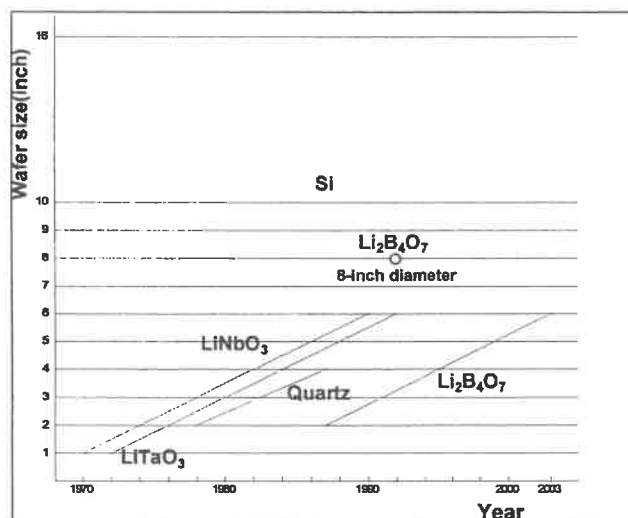


Figure 3 Single crystal wafer sizes and their trends towards large size

Al-Cu(0.5%)
Sputtering
Negative Photoresist Coating
TSMR-iN200(15cp)
Pre-Barking
95°C 120sec.
Deep-UV Exposure
2~4sec.
Post Exposure Barking
120°C 45sec.
Developing
AD-10
Post Barking
120°C 120sec.
Al-Cu(0.5%) Etching

Figure 4 $\text{Li}_2\text{B}_4\text{O}_7$ wafer lithography process

$\text{Li}_2\text{B}_4\text{O}_7$ Wafer Lithography Process

Alkali etching is adapted in photolithographic process for $\text{Li}_2\text{B}_4\text{O}_7$ wafer. The productive process of the developed lithography is shown in the mass production of the wafer for $\text{Li}_2\text{B}_4\text{O}_7$ in Figure 4. The negative type photoresist that the formation of a fine pattern of $0.3\mu\text{m}$ resolution is possible is used. Negative type photoresist (TSMR-iN200:Tokyo Ohka Kogyo) can control Line & Space. That is, if the exposure time is lengthened, Line & Space can be large, and becoming thin of the line of the photoresist at the etching time be corrected. Even if the pinhole is generated on the pattern, the negative type photoresist can avoid the IDT

short pattern. Post Exposure Baking (P.E.B.) process is indispensable, and the photoresist pattern changes depending on the condition so that the chemical amplification type negative photoresist may cause the negative type reaction after it exposes it. TSMR-IN200 is P.E.B.. Figure 5 shows after P.E.B. time of resist pattern. It changes into a lift-off process as a pattern formation method, and the alkali etching is used for the SAW device of the $\text{Li}_2\text{B}_4\text{O}_7$ substrate. Figure 6 is etching characteristics of the Al-Cu electrode film by the dip method. It is understood to be constant the etching speed, and to be able to control at time in 35°C temperature. This AL-4 is not seen as for the damage given to the TSMR-IN200 photoresist.

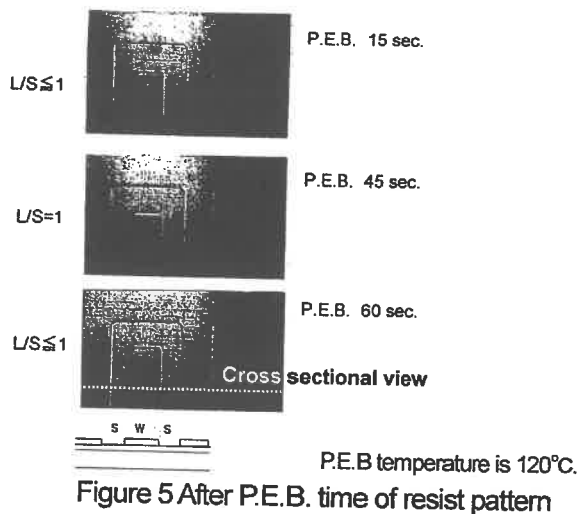


Figure 5 After P.E.B. time of resist pattern

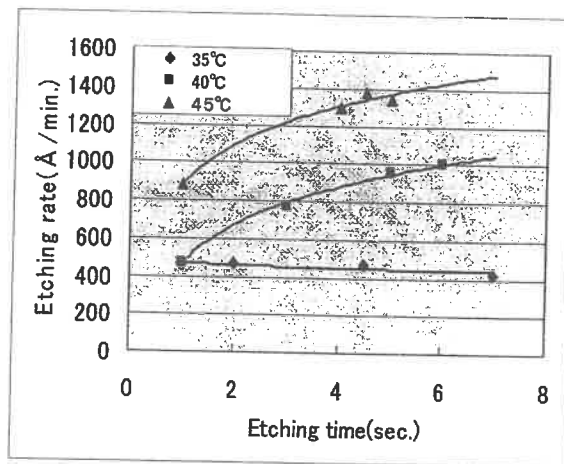


Figure 6 Alkali etching characteristics of AL-4

$\text{Li}_2\text{B}_4\text{O}_7$ material characteristics

It is examined that the electromechanical coupling factor improves the temperature characteristic of 1% and large $\text{Li}_2\text{B}_4\text{O}_7$ to ST-cut quartz. When the velocity of SAW is assumed to be V and the linear expansion coefficient in the direction where the substrate is assumed to be α , temperature coefficient of delay (TCD) of the structural SAW resonator shown in Figure 7 is given by the following expression.

$$TCD = \alpha - \frac{1}{V} \frac{\partial V}{\partial T} \quad \text{Here, it is } \alpha = \alpha_{11}(1 - \sin^2 \theta \sin^2 \varphi) + \alpha_{33} \sin^2 \theta \sin^2 \varphi.$$

In $\text{Li}_2\text{B}_4\text{O}_7$ of 45°X-Z (45,90,90), the linear expansion coefficient α becomes α_{33} (-4ppm/°C). Figure 8 shows the linear expansion curve in the direction of each crystal axis.

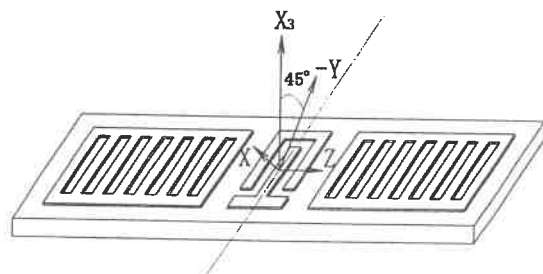


Figure 7 $\text{Li}_2\text{B}_4\text{O}_7$ SAW resonator

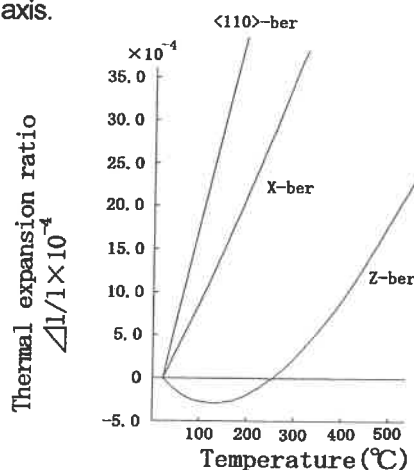


Figure 8 Thermal expansion curves of $\text{Li}_2\text{B}_4\text{O}_7$ (X-bar, Z-bar, and <110>-bar)

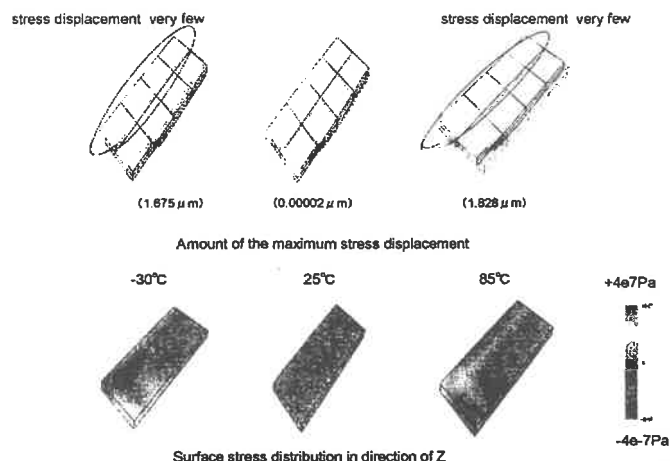


Figure 9 Anisotropic 3D heat stress FEM analysis result

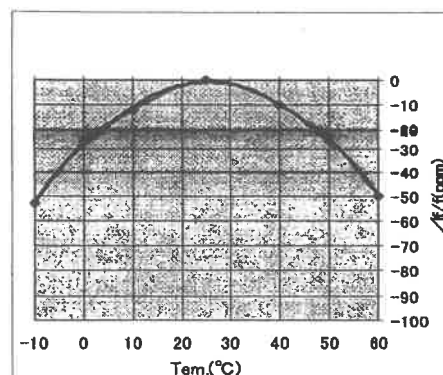


Figure10 Frequency temperature characteristics of New structural $\text{Li}_2\text{B}_4\text{O}_7$ SAW resonator

1/4 of model shown in Figure 7 is analyzed by using the anisotropic three dimensional heat stress FEM. Figure 9 shows the amount of the maximum displacement at each temperature. When it is $4.75 (Z \text{ direction length})/2 (Y \text{ direction length}) = 2.375$ compared with the vicinity, the amount of the surface displacement of SAW of $\text{Li}_2\text{B}_4\text{O}_7$ substrate is the smallest. Figure 10 is an obtained experiment value. The solid line is a temperature characteristic curve of ST-cut. Table 2 shows the performance index of various SAW resonators. This new structural $\text{Li}_2\text{B}_4\text{O}_7$ is the most excellent in $M_F=50.8$. (Range of temperature : 100°C)

Table 2 Performance index M_F of SAW Resonators

Substrate	$k^2(\%)$	γ	M_F
ST-cut Quartz	0.16	770	8.1
$128^\circ\text{rot Y-X LiNbO}_3$	5.5	22	3.2
X- $112^\circ\text{Y LiTaO}_3$	0.64	193	1.4
$\text{La}_3\text{Ga}_5\text{SiO}_{14}(0^\circ, 140^\circ, 25^\circ)$	0.36	342	7.3
$45^\circ\text{X-Z Li}_2\text{B}_4\text{O}_7$ (New Structure)	1	123	50.8

Conclusions

The following results are obtained.

1. It has been understood to obtain the temperature characteristic of ST-cut quartz about $\text{Li}_2\text{B}_4\text{O}_7$ SAW. It has been understood that the SAW chip shape is devised, and within the wide range of the temperature, there is a possibility
2. that high steady $\text{Li}_2\text{B}_4\text{O}_7$ SAW resonator can be realized. The mass production process of $\text{Li}_2\text{B}_4\text{O}_7$ SAW is established by combining the etching liquid of negative type photoresist and aluminum film (Al-Cu). The technique to have investigated this $\text{Li}_2\text{B}_4\text{O}_7$ can be applied as an algorithm to establish the manufacturing process to new functional materials.

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EXPERIMENTAL ANALYSIS OF DEPOSITION PROCESS OF LUBRICANT SURFACE BY ELECTRICAL DISCHARGE MACHINING WITH MOLYBDENUM DISULFIDE POWDER SUSPENDED IN WORKING OIL

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1. Introduction

Autonomous machines for space environment have been increasingly required since the construction of International Space Station began in 1998. A problem to be solved is lubrication between sliding parts such as joints. Solid lubricants, such as molybdenum disulfide (MoS_2), gold, silver or polytetrafluoroethylene (PTFE) are often used on the sliding parts exposed in the space environment instead of liquid ones because of evaporation and cosmic radiation. Spattering and sintering are usually used to deposit MoS_2 layer on a matrix for an exposure test in the space environment [1]. Parts to be modified deform by heat and their size is restricted in spattering process.

On the other hand, electrical discharge machining (EDM) has been used not only for removal process but also for deposit process [2]-[6]. In EDM with silicon powder suspended in working oil, surface roughness has been improved because of the dispersion of discharge [2] so that the friction coefficient has been decreased [7]. In addition, a surface has been modified by the diffusion of silicon [2]. Although a thick layer can be deposited by using a composite electrode [3], the production of a green compact takes much labor. When MoS_2 powder is suspended in the working oil, MoS_2 layer can be deposited on stainless steel during the finishing EDM process [8]. Many factors affect on the properties of deposited layer. The deposition process must be clear to improve the endurance of the lubricant layer.

In this paper, the deposition process is discussed and the influence of some machining conditions investigated through the friction test.

2. Experimental apparatus

A die-sinking electrical discharge machine (M35KC7G70 by Mitsubishi Electric Works) was used. Another working tank, which measures $280 \times 180 \times 130$ mm, with a pump was installed on the

table. The working fluid (EDF-K made by Nippon Oil Corp.) was filled over 50 mm from the workpiece surface and circulated with a flow rate of 19 l/min. The distance between the exit of a nozzle and the workpiece center was 10 mm.

The workpiece was washed in acetone for 10 min. with an ultrasonic washer before the evaluation process.

Table 1 shows the conditions in the friction test. In a reciprocating friction tester, a steel ball (JIS SUJ2/ASTM 52100) with a diameter of 7.9 mm was slid on a specimen driven with a crank mechanism. The vertical load can be varied by changing a weight mounted on the ball.

3. Observation of machining phenomena

In order to compare the phenomena in MoS_2 mixed EDM with those in Si powder mixed EDM, the discharge dispersion and the gap length between an electrode and a workpiece were measured.

The discharge dispersion was measured by the branched electric current method developed by Kunieda [9]. Fig. 1 shows a schematic view of the electrode unit. The current ratio between each path was calibrated beforehand. The dimensions of the electrode are $110 \times 10 \times 20$ mm.

Fig. 2 shows examples of the discharge dispersion under the conditions shown in Table 2. Though the discharge was always concentrated from the beginning of the machining in normal working oil as shown in Fig. (a), the discharge was uniformly dispersed not only at the beginning but also after 60 min. in the working oil mixed with MoS_2 powder as

Table 1 Conditions for friction test

Cycle time of reciprocating motion	1 s
Stroke	6.5 mm
Ambient temperature	18-28 °C
Humidity	13-16 %
Vertical load	0.22 N

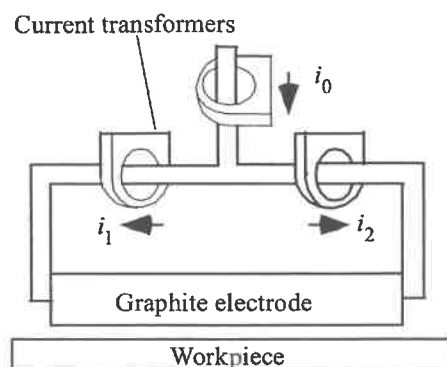


Fig. 1 Experimental setup of branched electric current method

Table 2 Machining conditions for dispersion test

Electrode polarity	Positive
Discharge current	6 A
Pulse duration	384 μ s
Pulse interval	192 μ s
Open gap voltage	80 V

shown in Figs. (b) and (c).

Then the gap length was measured under the conditions shown in Table 3. Fig. 3 shows the relationship between the gap length and powder concentrations. The gap length expanded with an increase of the powder concentration for each powder.

Because these phenomena are the same as those in EDM with Si powder, the main factors to make the surface smooth should be the discharge dispersion and the expansion of the gap length.

4. Lubrication performance

Then the reciprocating friction test in air was carried out. The lubricant layer was deposited on stainless steel under the conditions shown in Table 3

Table 3 Electrical conditions for MoS₂-mixed oil

Electrode polarity	Negative
Discharge current	2 A
Pulse duration	2 μ s
Pulse interval	8 μ s
Open gap voltage	320 V
Electrode	Cu ϕ 16 mm

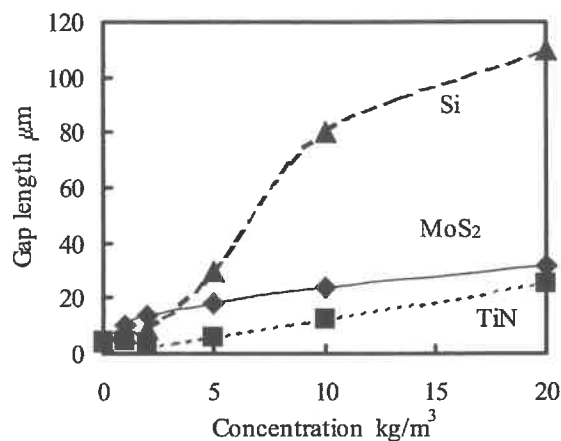


Fig. 3 Relationship between gap length and concentration of powders

and a powder concentration of 20 kg/m³. Fig. 4 shows examples of the reciprocating friction test. The friction coefficient was steeply increased after it exceeded over 0.3. Therefore, the lubrication duration was defined as the number of the reciprocating motion at which the friction coefficient exceeds 0.3.

Fig. 5 shows the dispersion of the lubrication duration along the flow of the working fluid. A low friction coefficient was sustained at the areas near the front edge and at the center of the deposit for long lubrication duration. The biased concentration in a gap between the electrode and the workpiece caused

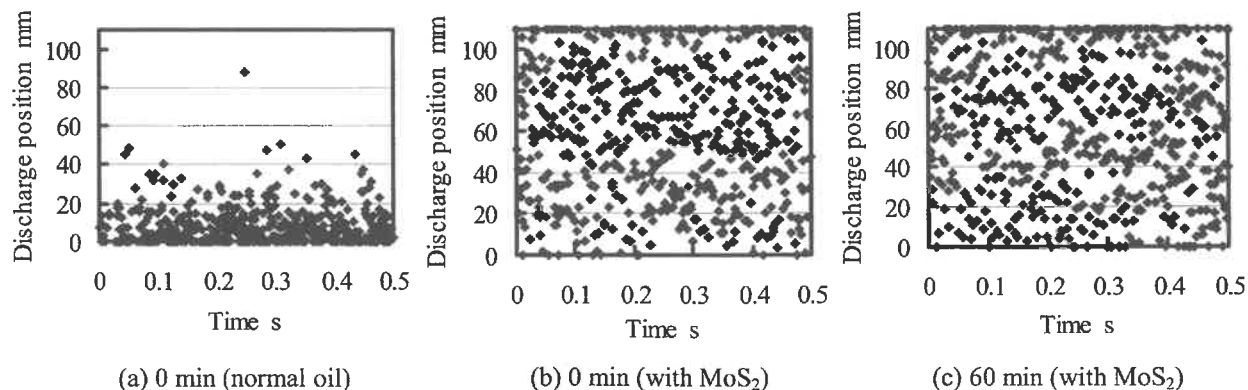


Fig. 2 Examples of discharge position

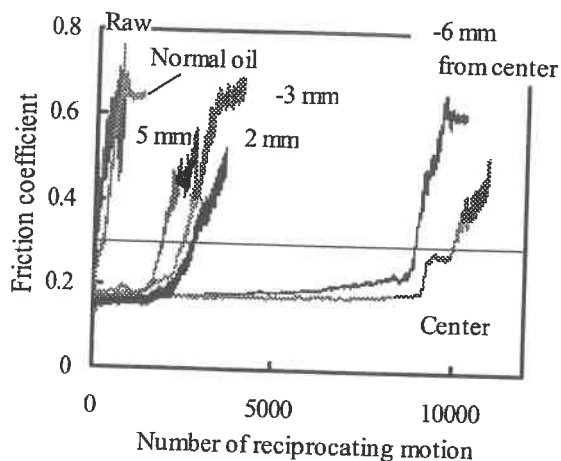


Fig. 4 Examples of reciprocating friction tests

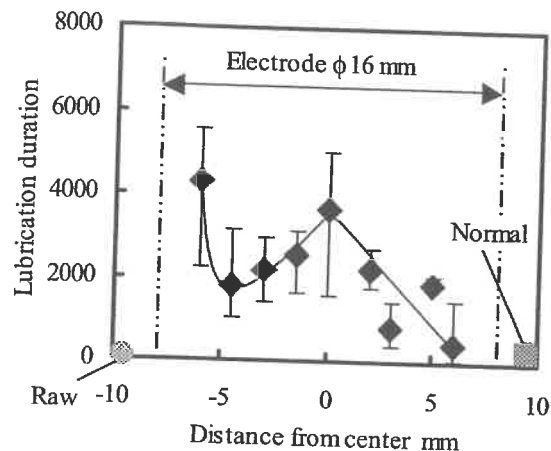


Fig. 5 Dispersion of lubrication duration in the case of rubbing parallel to stream

the unstable lubrication duration. No lubrication was observed on a raw or machined surface with normal oil. Fig. 6 shows an analysis result by electron probe micro analysis (EPMA). The concentrations of Mo and S were high at the boundary and lower stream side. However, the concentration of MoS_2 was low at C according to the result by the quantitative analysis. MoS_2 was decomposed or changed into another compound by the discharge heat.

Because this process is basically removal, deposited MoS_2 may be removed. Fig. 7 shows the relationship between the lubrication duration and machining time. The lubrication duration and roughness were saturated over 30 and 60 min, respectively.

5. Dominant properties on deposit

In order to find dominant properties and predict the deposit process, some metals were machined in the mixture of the working fluid and MoS_2 . Fig. 8 shows the influence of thermal properties. The solid circles and crosses indicate metals on which MoS_2

could be and could not be deposited, respectively. ZAS is a zinc alloy for stamping tools made by Mitsui Mining & Smelting Co. Because Sintered WC-Co on which MoS_2 could be deposited cannot be separated, WC and Co were independently indicated with the triangles. The product of the thermal conductivity and the melting point dominate the general removal process by EDM. After comparing the properties of the metals and MoS_2 , it, however, could be deposited on the metals with lower melting point than MoS_2 .

From the experimental results above, the deposition process was predicted as follows. When the MoS_2 concentration in the gap is appropriate, it adheres on the metal melted by the discharge heat. Then MoS_2 is caught with the re-solidified metal. Because the properties of the workpiece surface changes with a progress of the deposition, the deposit will be balanced with the removal and the concentration of MoS_2 will be saturated.

6. Conclusions

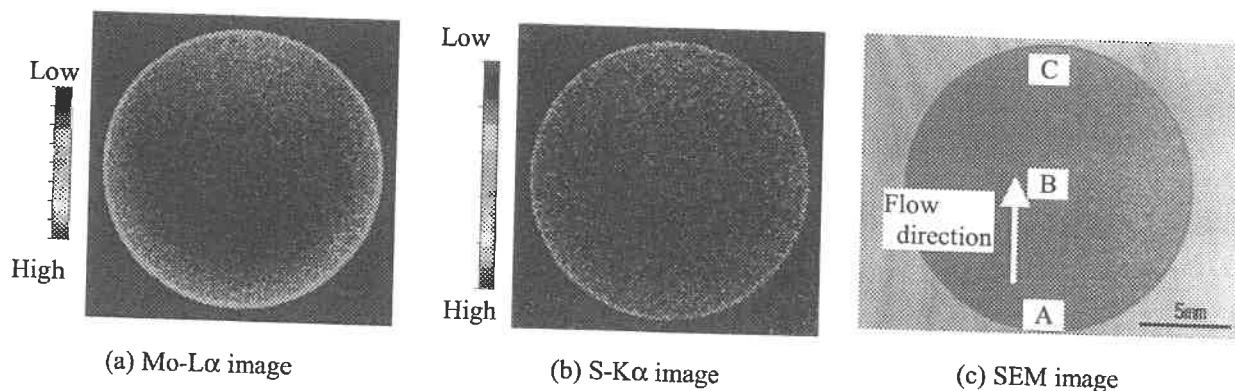


Fig. 6 Result of EPMA (color bars indicate concentration)

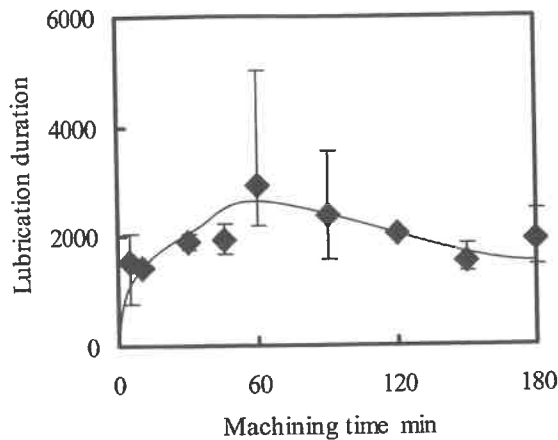


Fig. 7 Relationship between lubrication duration and machining time (center)

The phenomena during EDM with MoS_2 suspended in the working oil were observed. The conclusions can be drawn as follows.

- (1) MoS_2 could be deposited on metals which melting point is lower than MoS_2 .
- (2) The uneven concentration of MoS_2 caused the dispersion of the lubricant duration.
- (3) The expansion of the gap length caused the improvement of the roughness.

Acknowledgement

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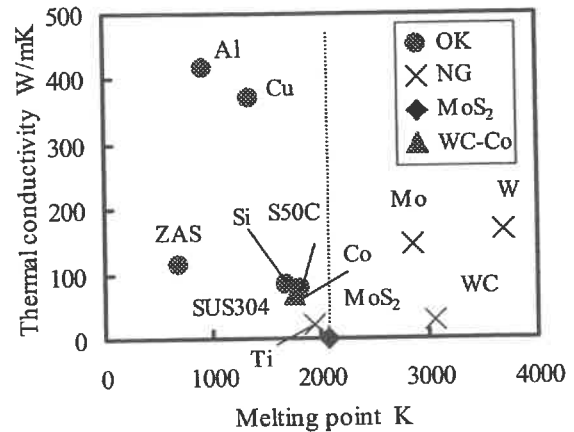


Fig. 8 Influence of thermal properties on deposit

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Investigations on Feasibility of Sub-micrometer Order Manufacturing Using Micro-EDM

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1. Introduction

In recent years, the need for the miniaturization of machining parts has been growing stronger in a wide range of industrial production technologies[1,2]. Other than conventional precision products such as cameras and watches, in the fields of medical devices, communication, data processing, space equipment, etc., miniaturized parts are becoming more and more valuable since they can provide more functions in limited space. This has led to growing demands for parts miniaturization. Especially with the increasing need for the development of the MEMS (Micro Electro Mechanical System), sub-micrometer or even nanometer parts are sought in various sectors. In order to meet these requirements, the study on sub-micrometer order manufacturing has become considerably important.

Since micro-EDM provides such advantages as the ability to manufacture complicated shapes with high accuracy, and process any conductive materials regardless of hardness, it has become one of the most important methods for manufacturing micrometer parts[3,4]. It is well known that rods larger than 5 μ m in diameter can easily be machined by micro-EDM. However, as mentioned above, rods smaller than 5 μ m or even sub-micrometer order in diameter are extensively required. Although numerous researches and development activities have focused on various applications of micro-EDM, no investigations have yet been conducted on the feasibility of sub-micrometer order manufacturing using micro-EDM.

In the present study, the smallest possible diameter that can be obtained for micro-rods was examined, and the factors affecting the limits for reducing diameter were investigated. Based on the investigation results, sub-micrometer machining was attempted.

2. Limits of minimum machinable size by conventional micro-EDM

The smallest possible size achievable for parts was investigated by machining tungsten rod under the machining conditions shown in Table 1. The WEDG method[5] was used to obtain tungsten micro-rods, and the pulse generator used was of the RC circuit type. Figure 1 shows the relationship between the target diameter and machined diameter, and Figure 2 shows a photo of the rod with the minimum diameter obtained from experiments. We found that the smallest diameter that can be obtained for tungsten rod was about 4 μ m. As for the target diameter of under 5 μ m, the diameter of machined rods was found to vary in each experiment and was inconsistent in the axial direction. Moreover, the predetermined rod length could not be obtained due to bending or breaking of the rod during the machining process. These experimental results revealed that sub-micrometer machining is not possible in the present situation.

3. Factors affecting limits of minimum machinable size

To determine the feasibility of sub-micrometer order manufacturing, we investigated the following factors which may affect the limits of minimum machinable size: 1) crystal defects preexistent in the workpiece material, 2) micro-cracks and holes in the damaged layer of

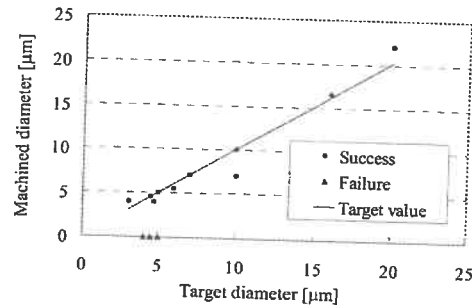


Fig.1 Relationship between target diameter and machined diameter

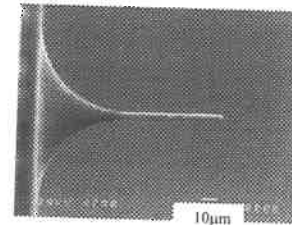


Fig.2 Minimum diameter rod under present conditions

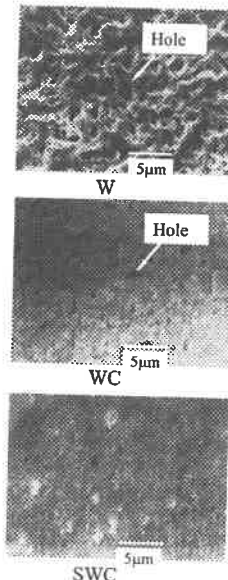


Fig. 3 Crystal structure of workpiece materials

Table 1 Machining conditions

	Open voltage[V]	Capacitance[pF]	Cutting depth[μm]
Roughing	110	3300	60
Medium roughing	90	1000	10
1st finishing	60	100	10
2nd finishing	40	10	5
Final finishing	40	Stray capacity(10)	2

the workpiece surface, 3) residual stress which may cause deformation of machined parts, 4) size of discharge crater, 5) positioning accuracy and thermal deformation of the micro-EDM machine. Since factors 1) to 3) are closely related to the material type, besides tungsten (W) which is often used in micro-EDM, cemented tungsten carbide(WC) and cemented tungsten carbide made of super fine particles (SWC) were also investigated in this study.

3.1 Crystal defects preexisting inside material

If the crystal defect existing inside the material is several μm , sub-micrometer machining is considered impossible. Crystal defects preexisting inside W, WC and SWC materials were observed by SEM (Figure 3) in this study. The crystal defect (hole) was found to be 2 to 3 μm for W, about 2 μm for WC, and almost no defects were found for SWC, suggesting that SWC is most suitable for sub-micrometer manufacturing among these three materials.

3.2 Measurement of damaged layer of machined surface

Since EDM machining applies thermal energy to melt the workpiece, damaged layers may be formed on the workpiece surface. This means that the minimum machinable size cannot be less than the thickness of the damaged layer. Here, W, WC and SWC rods were machined using the machining conditions shown in Table 1, and their damaged layers were observed by SEM. The thickness of the damaged layer for the W rod after roughing was found to be 5 μm , and this reduced to about 1 μm after finishing (Figure 4), indicating that the larger the machining energy, the thicker is the damaged layer. However, micro-cracks or holes of 1 μm were observed in the damaged layer even after the final finishing, indicating that sub-micrometer machining is difficult to realize. On the other hand, the damaged layers of the WC and SWC rods after roughing were about 1 μm (Figure 5), which is considerably thinner than the W rods, demonstrating that the damaged layer differs with the type of rod material. After the final finishing, the damaged layer was not clearly observed in the cross sections of the WC and SWC rods. From these results, it can be concluded that the damaged layer has negligible influence on sub-micrometer machining for WC and SWC.

3.3 Observation of discharge crater

The discharge crater has considerable influence on sub-micrometer machining. The machining of sizes smaller than the discharge crater is difficult. Here, the discharge craters of W, WC and SWC were measured under the roughing and final finishing conditions shown in Table 1. In order to observe the discharge crater generated by a single pulse discharge with a small discharge energy, the surface of the workpiece was mirror-finished, after which the discharge crater was observed by SEM. Figures 6 and 7 show photos of the discharge craters on W, WC and SWC after roughing and final finishing respectively. The size of the discharge crater obtained by roughing was 11.5 μm for W, 12 μm for WC and 12 μm for SWC, indicating no significant difference with respect to the material type. The size of the discharge crater obtained by final finishing was about 2 μm for all materials. From this, it was concluded that the machining of sizes smaller than 2 μm is difficult using the micro-EDM machine adopted in this study, because the discharge energy in the final finishing shown in Table 1 is minimum for the machine used.

Normally, to obtain higher material removal rates in micro-EDM, the workpiece is usually set as the anode and the wire electrode as the cathode (straight polarity machining). This is because the discharge energy distributed

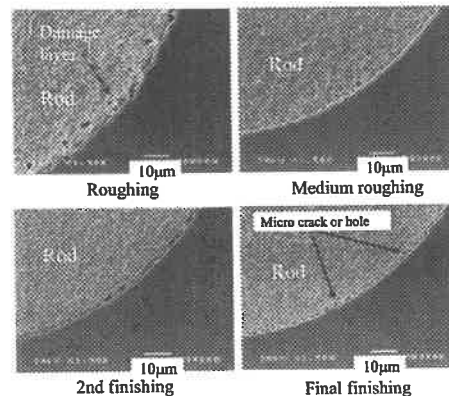


Fig. 4 Damaged layer of W rod after electrical discharge machining

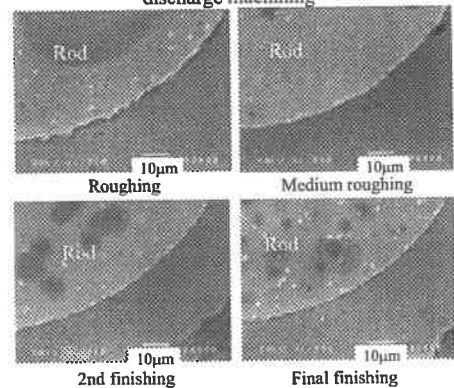


Fig. 5 Damaged layer of WC rod after discharge machining

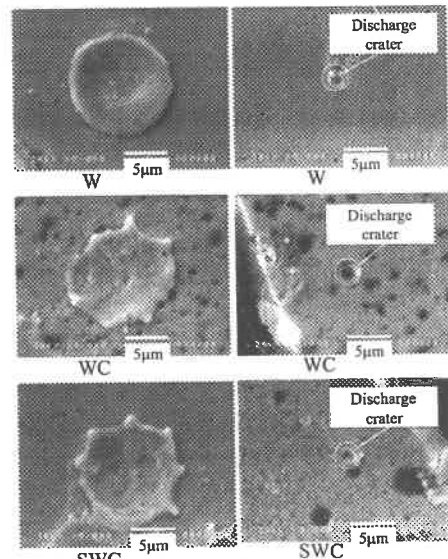


Fig. 6 Discharge crater after roughing

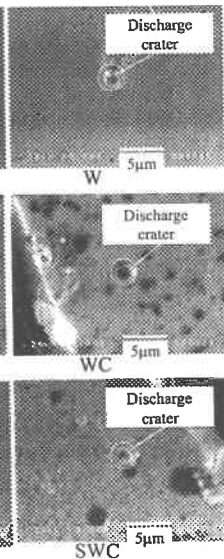


Fig. 7 Discharge crater after final finishing

to the anode is normally greater than that to the cathode [6]. This fact indicates that the discharge crater generated on the cathode surface is smaller than that on the anode. Based on this, in this study, the discharge crater on the cathode workpiece (SWC) surface obtained by reversed polarity machining was observed. Compared with the discharge crater generated on the anode workpiece (SWC) surface in straight polarity machining (Figure 7), the profile of the discharge crater shown in Figure 8 is not clear, and the diameter is slightly smaller. The size of the discharge crater under the final finishing conditions with the reversed polarity was about $0.8\mu\text{m}$ for W and $1\mu\text{m}$ for SWC.

3.4 Influence of residual stress

The tungsten rod is susceptible to deformation during the machining process due to residual stress which may preexist because tungsten rod is manufactured by drawing. The residual stress may also be generated by the thermal load of discharge during the EDM process itself. A rod measuring $17\mu\text{m}$ in diameter and $1000\mu\text{m}$ in length was machined using W, WC and SWC by roughing and finishing. Since the residual stress is difficult to measure directly, the deformation of the rod after machining was measured in order to estimate the influence of residual stress as shown in Figure 9. The measured results in Table 2 indicate that the deformation of W is larger than that of WC and SWC, demonstrating that W is not suitable for sub-micrometer machining. Deformation was small for the WC and SWC rods because those rods are not produced by drawing but by sintering, and the Young's Modulus of these materials is higher than that of tungsten.

3.5 Influence of positioning accuracy of micro-EDM machine

The positioning accuracy of the micro-EDM machine was classified into three types in this study: home positioning accuracy, repetition accuracy and thermal deformation. The micro-EDM machine used in this study had three axes; X, Y and Z. The home positioning accuracy and repetition accuracy were measured for the X axis. An electric micrometer was used and its sensor head was set on the X axis. The home positioning accuracy was measured after 20 times of returning to the machining origin from points $X = -19.0$ and $Y = -3.5$. The repetition accuracy was measured after 20 repetitions of the reciprocating motion of the X axis for a distance of 5mm . The measurement results of the home positioning accuracy and repetition accuracy are shown in Figure 10. The horizontal axis represents the sequence of experiments, and the vertical axis represents the positioning error from the starting point. An error of $\pm 0.3\mu\text{m}$ was found for the home positioning accuracy. As for the repetition accuracy, a maximum error of $0.5\mu\text{m}$ was found. Considering the home positioning accuracy and repetition accuracy together, the positioning accuracy of the machine was over $1\mu\text{m}$, suggesting that this may be one of the reasons why the diameter of machined rods looked varied when the target diameter was under $5\mu\text{m}$.

The influence of thermal deformation on the positioning accuracy was indirectly investigated by measuring machined shape error. That is, since it took about one hour to machine a rod of $16\mu\text{m}$ in diameter and $1000\mu\text{m}$ in length, fluctuation of the rod diameter along the rod axis was considered to be the influence of thermal deformation. Here, it should be noted that this error may include the influence of misalignment of the rotation axis in parallel with the feeding direction. Figure 11 shows the experimental results. The maximum deviation was $2.4\mu\text{m}$. These results indicate that the influence of thermal deformation on geometrical accuracy cannot be ignored in sub-micrometer machining.

4. Attempts of sub-micrometer machining using SWC

From all the results obtained thus far, it can be concluded that sub-micrometer machining using any of the materials W, WC, and SWC is difficult due to insufficient positioning accuracy of the micro-EDM machine used. Disregarding positioning accuracy, WC and SWC are relatively better than W in terms of crystal structure, damaged layer, and influence of residual stress. In particular, SWC is more suitable than WC because both

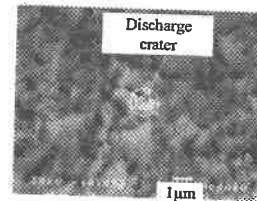


Fig. 8 Discharge crater generated on cathode workpiece (SWC) surface with reversed polarity after final finishing

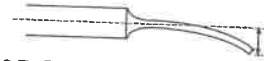


Fig. 9 Deformation due to residual stress

Table 2 Measurement results of deformation

	Roughing	Final finishing
W	$72[\mu\text{m}]$	$5[\mu\text{m}]$
WC	$2[\mu\text{m}]$	$2[\mu\text{m}]$
SWC	$2.5[\mu\text{m}]$	$1[\mu\text{m}]$

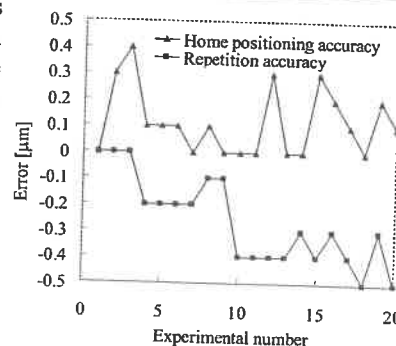


Fig. 10 Positioning accuracy

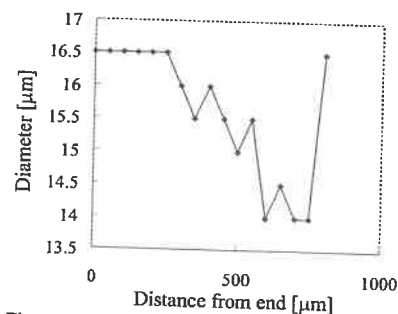


Fig. 11 Distribution of diameter in axial direction

crystal grain size and crystal defect size are smaller. Based on these results, sub-micrometer machining using SWC was attempted with both straight and reversed polarity. The machining conditions are shown in Table 3. The minimum rod obtained from straight polarity was about 2.8 μ m. Figure 12 shows its photo. Next, to achieve sub-micrometer machining, SWC was machined with reversed polarity, but only in the final finishing stage shown in Table 3, because the material removal rate with reversed polarity is considerably lower than that of straight polarity. Unfortunately, use of reversed polarity did not provide success of sub-micrometer order machining.

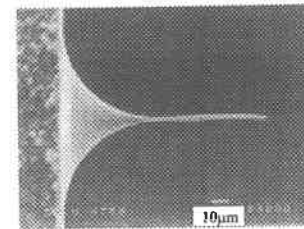


Fig.12 Minimum diameter SWC rod obtained using straight polarity

The minimum diameter obtained by using reversed polarity was almost the same as that using straight polarity. The machining time was about eight times longer than straight polarity as shown in Figure 13. Hence, the main reason for failure to obtain sub-micrometer machining in reversed polarity was attributed to thermal deformation. The machining surfaces obtained using straight polarity and reversed polarity were also observed (Figure 14) in the experiments. The surface obtained by reversed polarity was found to be smoother than that using straight polarity. This result coincides with the result where the discharge crater generated on the workpiece surface with reversed polarity is smaller than that of straight polarity, indicating that reversed polarity machining has potentials for achieving sub-micrometer machining.

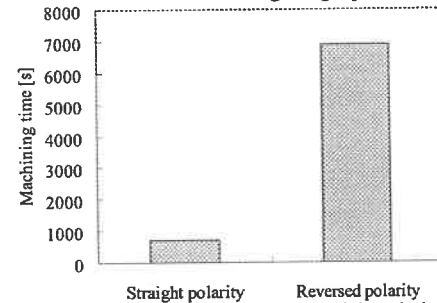


Fig.13 Machining time using straight polarity and reversed polarity

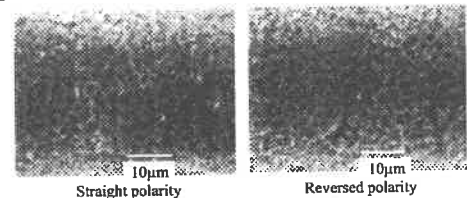


Fig. 14 Machined surfaces

5. Conclusions

From investigations on factors which may affect the smallest possible rod diameter that can be obtained by micro-EDM, it was found that, 1) positioning accuracy and the smallest discharge energy of the micro-EDM machine used are not suitable for sub-micrometer machining. 2) Disregarding the positioning accuracy and smallest discharge energy of the micro-EDM machine, WC and SWC are relatively better than W in terms of crystal structure, damaged layer, and influence of residual stress. In particular, SWC is more suitable than WC because both crystal grain size and crystal defect size are smaller. 3) For polarity, the discharge crater was smaller and the finished surface was smoother with reversed polarity than with straight polarity. Based on the investigation results above, sub-micrometer machining using SWC was attempted, and the smallest possible diameter obtained was about 2.8 μ m. Although we were not able to achieve sub-micrometer manufacturing in this study, the findings will provide useful strategies for future sub-micro-EDM techniques.

Table 3 Machining conditions

	Open voltage[V]	Capacitance[pF]	Cutting depth[μ m]
Roughing	110	3300	130
Medium roughing	90	1000	10
1st finishing	90	100	5
2nd finishing	80	10	2
3rd finishing	60	Stray capacity(10)	1
4th finishing	50	Stray capacity(10)	0.5
Final finishing	40	Stray capacity(10)	0.2
Final finishing	40	Stray capacity(10)	0.2
Final finishing	40	Stray capacity(10)	0.2
Final finishing	40	Stray capacity(10)	0.2

Acknowledgements

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SILICON ON INSULATOR (SOI) WAFER POLISHING USING MAGNETORHEOLOGICAL FINISHING (MRF) TECHNOLOGY

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1. INTRODUCTION

The specifications for the top layer of silicon on SOI (silicon-on-insulator) wafers continue to grow tighter. As reported in the December 2001 edition of the International Technology Roadmap for Semiconductor (ITRS-2001), the requirements for absolute thickness and thickness variation of the silicon layer are pushing the limits of both the current metrology and fabrication processes [1]. Traditional manufacturing processes cannot meet the proposed thickness specification (T_{si}) of 20-50 nm and associated thickness variation of $\pm 5\%$ of the T_{si} . The ITRS-2001 predicts that these requirements will become increasingly difficult to meet as absolute thickness specification (T_{si}) drops to 20 nm with an allowable thickness variation as low as 2 nm peak-to-valley. Magnetorheological finishing (MRF) is uniquely able to target the absolute thickness of the silicon layer while significantly, and simultaneously, improving its thickness variation. This paper will describe MRF and its viability as a manufacturing solution, allowing wafer fabricators to reach the current, and future, SOI specifications. Initial results obtained using the MRF process on SOI wafers will be demonstrated.

2. MRF PROCESS

Magnetorheological finishing (MRF) is a deterministic, final figuring process developed to address many of the limitations encountered in traditional polishing. The three key components to deterministic polishing processes are 1) a stable polishing tool, 2) repeatable and reproducible metrology data, and 3) software to de-convolve the metrology error map and the removal rate of the polishing tool to obtain an accurate dwell schedule. The MRF polishing tool is based on a magnetorheological fluid whose viscosity is changed significantly in the presence of a magnetic field. Using MR fluid as a polishing tool presents significant advantages compared to other sub-aperture polishing approaches, including the ability to adjust removal rate and the insensitivity to z-axis positioning [2]. There are several parameters which can be controlled to increase or decrease the removal rate of the polishing tool (e.g., magnetic field, depth into fluid, viscosity of fluid, and wheel speed.) Figure 1 displays an SOI wafer while being polished with the MRF.

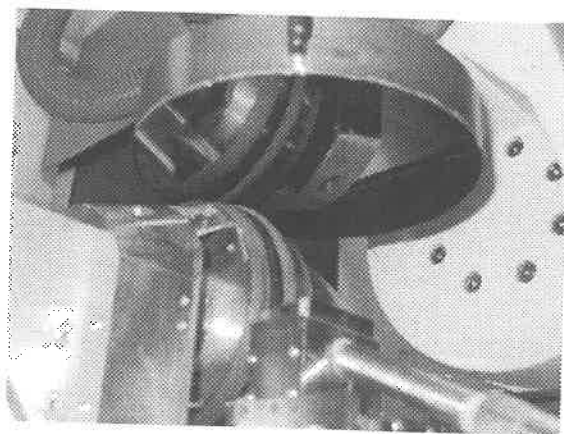


Figure 1. Image of an SOI wafer being polished with the MRF process. The wafer is held with vacuum on a New Way Precision chuck [3].

The MRF process has been successfully implemented in the fabrication process of high quality lithography optics and many other applications [4]. While achieving less than 0.5 nm rms micro-roughness, surface figure error below 2 nm rms is simultaneously reached on various types of optical glass (such as BK7 and SiO₂) and single crystal materials (such as silicon and CaF₂) [5]. Typically for these types of optical components, the primary advantages of MRF are associated with its very deterministic nature and ability to polish a wide range of geometries (flat, spheres, aspheres, cylinders, etc.) to an unprecedented level of precision in a fast and cost-effective manner. For most optical applications, MRF is typically used to shape a lens based on reflected or transmitted wavefront metrology. In the case of SOI wafers, on the other hand, it is the thin film of Si on top of a SiO₂ layer (itself on a silicon layer) for which thickness and thickness uniformity needs to be improved.

3. SOI THICKNESS MEASUREMENTS

Standard methods to measure the thickness and thickness variation of the top silicon layer of SOI wafers are reflectometry and ellipsometry [6]. For work done in this research, thickness and thickness variation measurements were made using spectral reflectometry with an ADE AcuMap II tool [7]. Using this equipment, measurements were made over the entire surface and sampled every millimeter, producing over 20,000 data points for a 200 mm diameter wafer in under a few minutes. Repeatability of these measurements was estimated to be approximately 0.5 Å variability. While techniques such as ellipsometry may be even more accurate, the measurement cycle time is much higher and the number of measurements on the surface are much fewer (typically 125 points per surface). At this low data density, some surface errors can be easily missed.

4. MRF on SOI PROCESS

The final goal of this research was to produce an SOI wafer with a silicon thickness of 20-50 nm and associated thickness variation less than 5% of the nominal thickness. The goal consists of two separate tasks: nominal thickness reduction and thickness variation correction. The deterministic nature of the MRF process can very effectively address both tasks. A precise CNC machine was used to position the SOI wafer into the MR fluid ribbon. The wafer is held onto a porous ceramic chuck using vacuum (see Figure 1). In most cases, SOI wafers have an initial silicon thickness ranging from $T_{si} = 50$ -100 nm and a uniformity of 5% of the T_{si} . With initial conditions such as these, one approach to meet the desired SOI specification is to divide the MRF polishing into two iterations. The first iteration is designed to bring the thickness close to the desired thickness, but with enough material remaining to perform a second iteration. The first iteration can be conducted with minimal metrology performed on the wafer. If the process used to fabricate the wafer produces repeatable thickness values, then initial metrology would not be necessary. The second iteration is designed to reduce the thickness to the final desired thickness and simultaneously correct the thickness variation. This second iteration, on the other hand, requires that metrology be performed prior to MRF. For the initial process investigation and development, the two tasks were conducted separately.

5. INITIAL RESULTS

The first step was to determine how accurately the MRF process could achieve a given thickness. As mentioned earlier, the first polishing iteration was performed to reduce the silicon thickness to just above the desired thickness while leaving enough material for the second iteration. Figure 2 displays an example of radial average thickness results before and after MRF polishing of a 200 mm diameter wafer.

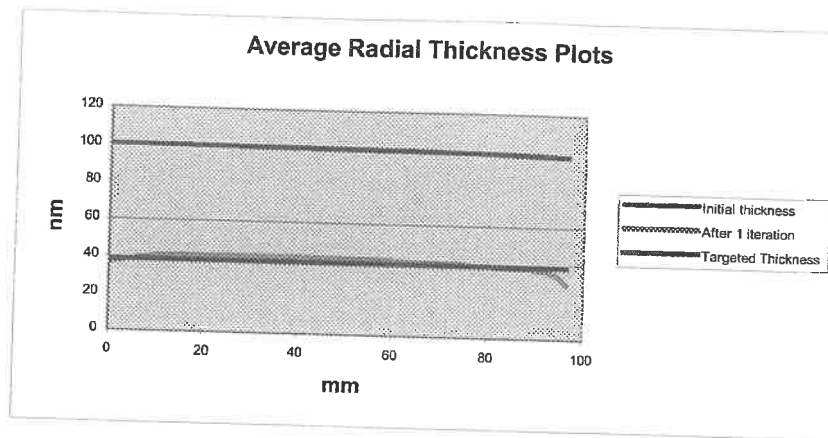
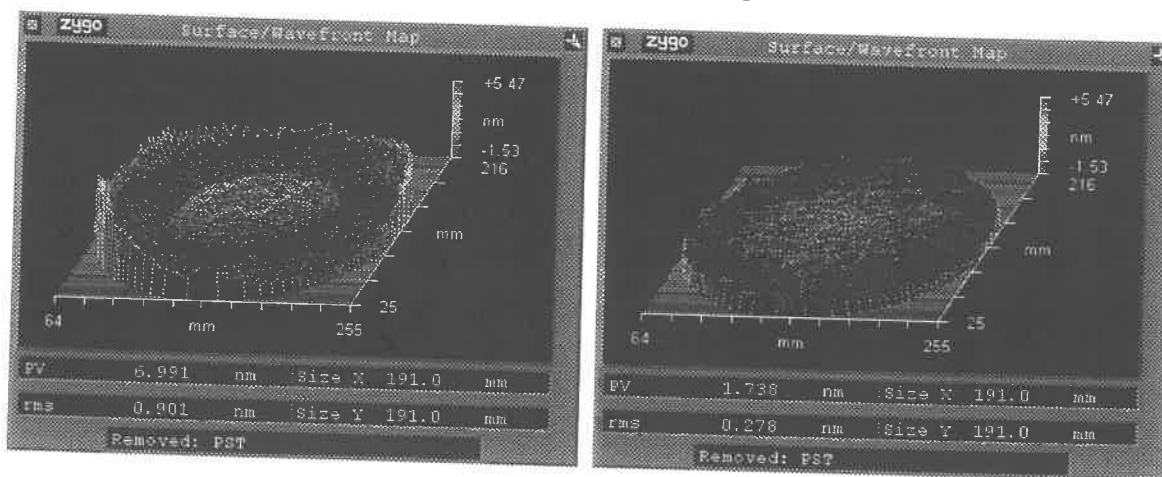


Figure 2. Silicon thickness measurements (with 2.5 mm edge exclusion) made before and after a MRF thickness reduction run.

The initial average thickness of the silicon layer was 99 nm as measured with the AcuMap II. The goal was to remove three-fourths of the material required to reach the 20 nm thickness target. With no metrology input, the MRF was programmed to uniformly remove a total of 55 nm from the “thinnest” area of the silicon layer. This set the target thickness to 38 nm. The resulting average silicon thickness following 37 minutes of MRF polishing was 39 nm. For this iteration, the error in thickness reduction was less than 2%. This result demonstrates the ability of the MRF process to very precisely target and reach a defined thickness.

The second task was to reduce the thickness variation of the silicon layer. The approach used to correct the thickness uniformity of the silicon layer was to change the process parameters to reduce the typical removal rate of the MRF process to a lesser removal rate. Figure 3 displays the thickness variation results of the silicon layer before and after MRF processing.



(a)

(b)

Figure 3. Silicon thickness variation measurements made (a) before and (b) after MRF processing with reduced removal rate.

The results show that the thickness variation was reduced from 7 nm to below 2 nm peak-to-valley, and even more impressively, achieving less than 0.3 nm rms even with the high-density measurement made with the AcuMap II. The total cycle time of the polishing run was 46 minutes. Work is currently being performed to achieve similar or improved uniformity results using more aggressive removal

settings. Initial results show that the uniformity correction cycle time can be reduced to less than 5 minutes.

6. CONCLUSION AND RELEVANT WORK

From the initial series of experiments performed using the MRF process on SOI wafers, the results show that the MRF process can successfully achieve the desired silicon thickness of 20-50 nm while reducing the thickness variation to less than 5% of the T_{Si} . Currently, work is being performed to combine the two separate tasks (reducing silicon thickness and correcting thickness variation) on a single operation.

In parallel, work is being performed on "thick" SOI and prime silicon wafers. Thick SOI wafers typically have a silicon layer thickness of several microns and again a uniformity of 5% of the T_{Si} . The approach to implement the MRF process for this type of wafer is different from the "thin" SOI process presented here. For this application, the parameters are adjusted to produce a more aggressive removal rate, as the amount of desired material removal is on the order of microns, versus ~50-70 nm removed for thin SOI wafers [8].

Investigations have also been performed to quantify many other additional important aspects of polishing Si or SOI wafers, including roughness, cycle time, contamination, process integration issues, etc. Initial findings have verified that MRF meets or exceeds industry standards in these areas and is a viable manufacturing solution for SOI wafer fabrication [8].

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Surface Modification of EDMed Surface by Wide-area Electron Beam Irradiation

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1. Introduction

In general, the surface of metal molds is finished by hand lapping after milling and/or EDM, in order to attain high shape accuracy and small surface roughness without cracks and heat affected layer. This process takes a lot of time and needs special technical skills. Therefore, a high efficient process is strongly required.

In this study, a new finishing process for metal molds by wide-area electron beam (EB) irradiation is proposed. In the wide-area EB irradiation equipment, EB with high energy density can be irradiated without concentrating the beam in this method. Therefore, wide-area EB with a maximum diameter of 60mm can be used for melting or evaporating metal surface. Using this system, the possibility for high efficient finishing of metal molds is experimentally discussed.

2. Experimental Procedures

Figure 1 illustrates the EB irradiation equipment and the generation mechanism of wide-area EB⁽¹⁾⁽²⁾. Differently from general EB irradiation performed in a vacuum, an argon gas of about $(5-15) \cdot 10^{-2}$ Pa is beforehand mixed in the chamber. At first, a magnetic field is generated by the solenoid coil set on the outer side of chamber. At the moment when the magnetic field takes a maximum intensity, pulse voltage is loaded to the anode. In the chamber, The electrons generated by Penning effect start to move towards the anode. Simultaneously, Lorentz force acts on the electrons, and so they move spirally. Next, argon gas atoms are ionized by the repetitious collision with electrons, and plasma is generated near anode. When the plasma intensity takes a maximum, pulse voltage is applied to the cathode, and the electrons are accelerated by high electric field due to electric double layer formed near cathode. Then, EB with high energy density is irradiated to workpiece surface. Moreover, the plasma is effective to make the life of EB long. That is, by passing through plasma region, the Coulomb's force between electrons is decreased and the straightness of EB can be improved. In this system,

the EB irradiation is carried out in a series of pulses. By the above-mentioned mechanism, high energy density EB can be produced without concentrating the beam. Therefore, EB with a maximum diameter of 60mm has a high energy density enough to melt or evaporate metal surface.

The EB irradiation time of one pulse is only 2-3 μ s, and the pulse frequency is set to 0.2Hz. The diameter of EB is 60mm as mentioned above, and the acceleration voltage is 30keV. Metal mold steel (NAK80, Daido Steel Co. Ltd.) is prepared as the workpiece material. the surface is beforehand EDMed using copper cylindrical electrode of 8mm in diameter, and the surface roughness is about 6 μ mRz.

3. Results and Discussions

3.1 Effects of energy density

At first, an optimum condition for smoothing surface is investigated by changing the energy density of the beam. Figure 2 shows SEM micrographs of the EB irradiated surface for various energy densities of electron beam. EDMed surface is also shown for comparison. Number of irradiation is 30 times. Under small energy density condition of 1.4J/cm², some melted parts can be observed on the surface. In the case of 2.1J/cm², the surface melting are confirmed

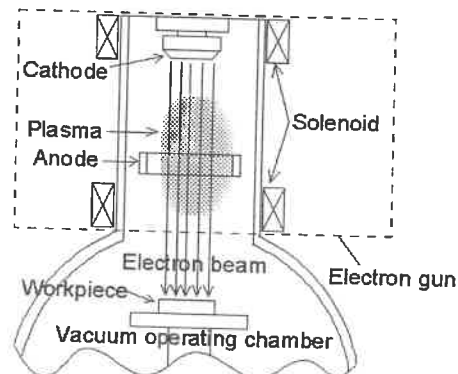


Fig.1 Wide-area electron beam irradiation

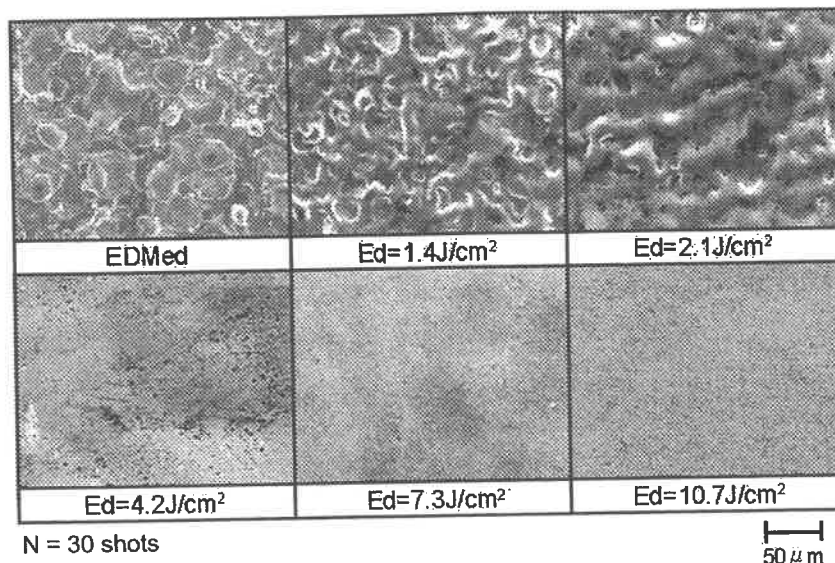


Fig.2 Wide-area EB irradiated surfaces for various energy densities

obviously and it seems to become smoother. Furthermore, under larger energy density condition, the state of surface completely differs from the EDMed surface before EB irradiation. **Figure 3** represents graphically the variations of surface roughness and the glossiness with the energy density. The measurements of glossiness were carried out in accordance with JIS specifications (Z8741). In this case, the glossiness of the perfect mirror surface is just 1000. As shown in the figure, the surface roughness becomes smaller with increasing the energy density, and takes a minimum value of $0.7\mu\text{mRz}$ at $6.7\text{J}/\text{cm}^2$. However, excessive energy density makes it a little worse. On the other hand, the glossiness becomes higher with increase of energy density, and corresponds well to the change in the surface roughness.

3.2 Effects of number of irradiation

Next, the effect of number of EB irradiation is discussed, and the surface roughness became smaller with number of EB irradiation. Sufficient smoothing of the surface might be possible by relatively large energy density of EB or large number of EB irradiation. In other words, it is guessed that the roughness depends on the total energy density, which is the product of the energy density and the number of irradiation. Then, the variations of surface roughness with the number of irradiation are compared for two different energy density conditions. **Figure 4** shows the variations of surface roughness in the cases of $4.2\text{J}/\text{cm}^2$ and $7.3\text{J}/\text{cm}^2$. The surface roughness decreases with an increase of number of irradiation, and takes a minimum at the total energy density of 200-300 J/cm^2 under both energy density conditions. However, the minimum value of

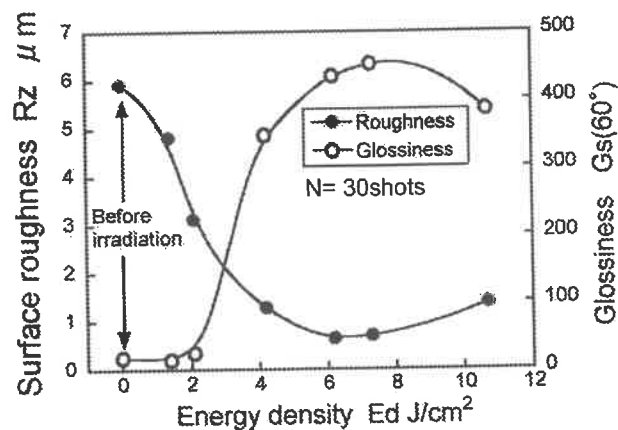


Fig.3 Variations of surface roughness and glossiness with energy density

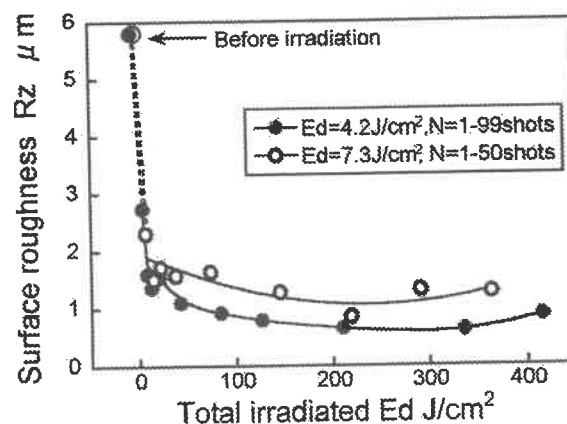


Fig.4 Variations of surface roughness with total energy density

surface roughness in the case of $4.2\text{J}/\text{cm}^2$ is smaller than that of $7.3\text{J}/\text{cm}^2$. Therefore, sufficient smoothing can be attained under the condition of small energy density and large number of irradiation.

3.3 Effect of surface roughness before irradiation

The EDMed surface of about $6\mu\text{mRz}$ in roughness was discussed above. If the sufficient smoothing of the surface with larger roughness is also possible, the efficiency of finishing process for metal mold might become higher. Then, the effect of the surface roughness before EB irradiation on the surface roughness after the irradiation was discussed. Figure 5 represents the relationship between the surface roughness before and after EB irradiation. The EB irradiating conditions were optimum condition made clear in the previous paragraph ($7.3\text{J}/\text{cm}^2$, 30shots). The surface roughness of EB irradiated surface is influenced by the surface roughness of EDMed. And the surface roughness after irradiation becomes about

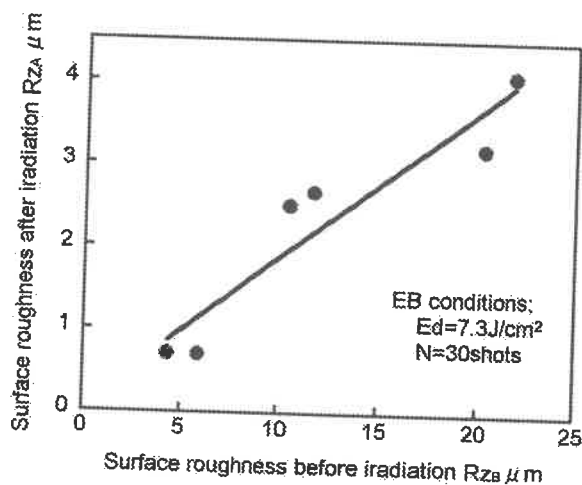


Fig.5 Relationship between surface roughness before and after EB irradiation

one fifth of that after irradiation under the same EB irradiating condition. Therefore it can be understood that the surface roughness by this finishing process greatly depends on the EDMed surface roughness before process.

3.4 Smoothing mechanism

SEM micrographs in Figure 6 are the cross sections of EDMed surface, EB irradiated one after EDMing, and EB irradiated one after grinding, in the order from the left. In the case of EDMed surface, a resolidified layer (white layer) in which, once melted material with high temperature due to electrical discharge is solidified again, can be observed clearly, and the undulation of surface is large. On the other hand, the thickness of white layer decreases, and the undulation becomes small, in the case of EB irradiated surface after EDMing. Moreover, there seems to be no white layer on the EB irradiated surface after grinding. Therefore, the white layer, which can be observed on EB irradiated surface after EDMing is formed by not EB irradiation but by EDM. In this EB irradiation system, EB irradiation time of one pulse is so short as $2\text{-}3\mu\text{s}$, and the energy density is relatively small as $10^6\text{W}/\text{cm}^2$. Therefore, it is guessed that the effect of heat conduction on the surface is so small, and the electron beam does not go into inside material, and the material removal by melting and evaporation occurs just near the surface.

3.5 Improvement of corrosion resistance

Figure 7 shows the wide-area EB irradiated sample and non irradiated one left for 1 year in the atmosphere. Circular part is EDMed surface and the surrounding is ground one. In the case of non EB irradiated workpiece, the surface extensively rusted. On the other hand, there is no rust, and glossy surface could be kept. Therefore, the corrosion resistance can be improved very much by wide-area EB irradiation.

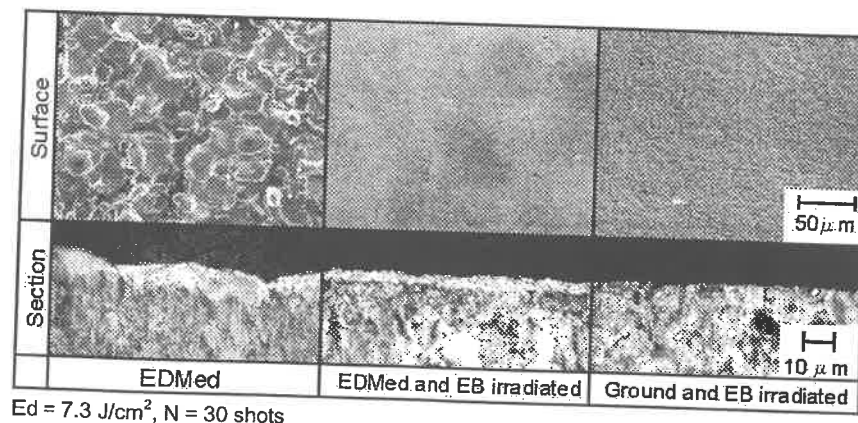


Fig.6 Cross sections of EDMed surface and wide-area EB irradiated surfaces

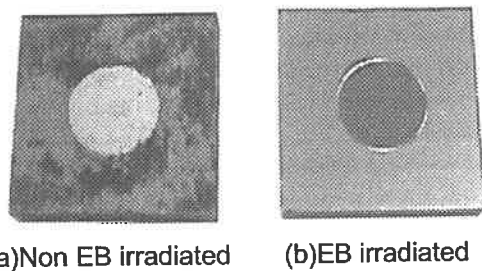


Fig.7 Workpieces left for 1 year in atmosphere

3.6 Smoothing of tilting surface

Here, the smoothing of tilting surfaces is tried for actual use, since metal molds usually have many tilting curved surfaces. Then, as shown in Figure 8, when the tilting angle of surface for EB irradiation direction θ varies, the surface roughness is measured. If the surface roughness after EB irradiation is decided by only the total energy density Ed , the surface roughness will be the same one with energy density of $Ed_0 \cdot \cos\theta$. Experimental results are shown in Figure 9. The dotted line in the graph is the expected one by $Ed_0 \cdot \cos\theta$. In the case of small tilting angle, as a matter of course, the surface roughness decreases to $0.7\mu\text{mRz}$ and it is almost the same as expected. On the other hand, the experimental surface roughness is much smaller than the expected one in the case of large tilting angle. For example, when the tilting angle is 60 degrees, the surface roughness is about $1\mu\text{mRz}$, while the expected one is about $2\mu\text{mRz}$. Furthermore, even if the tilting angle is close to 90 degrees, the surface roughness can be improved. The results indicated that in the case of small simple shaped mold consisting of relatively small tilting surfaces, the smoothing of the whole surface is possible in a few minutes without moving and tilting the mold.

4. Conclusions

- (1) The roughness of wide-area EB irradiated surface becomes smaller with increasing the energy density of beam and the number of irradiation. Under a proper condition, the surface roughness decreases to about $0.7\mu\text{mRz}$ in just a few minutes
- (2) In this smoothing process by wide-area EB irradiation, the material removal by melting and evaporation occurs near the surface and resolidified layer does not form.
- (3) The surface roughness of EB irradiated surface is greatly influenced by the surface roughness of EDMed before EB irradiation.

Wide-area Electron Beam

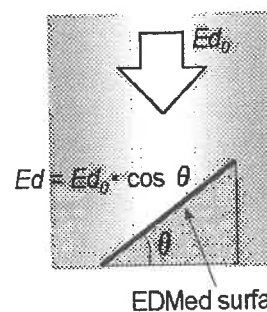


Fig.8 Wide-area EB irradiation to tilting surface

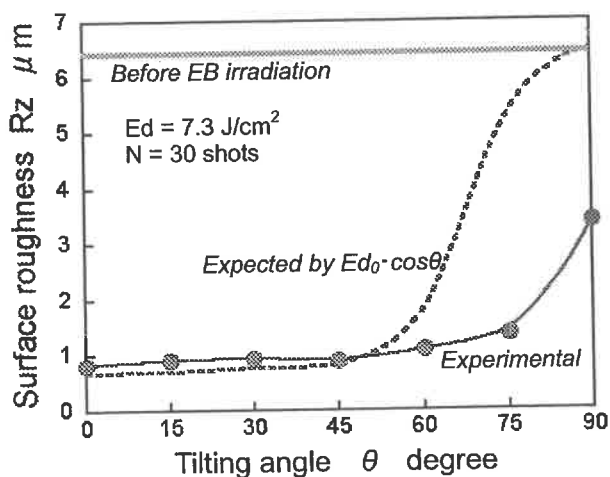


Fig.9 Variations of surface roughness with surface tilting angle

- (4) Even if the tilting angle of surface for EB irradiating direction is close to 90 degrees, the surface roughness could be well improved.
- (5) Wide-area electron beam irradiation has a possibility to be a high efficient finishing process for metal molds.

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SUPERFINISHING OF ALLOY STEELS USING MAGNETIC ABRASIVE FINISHING PROCESS

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Abstract

Magnetic abrasive finishing (MAF) process is the one in which material is removed in such a way that surface finishing and deburring are performed simultaneously with the applied magnetic field in the finishing zone. The mechanism of superfinishing in any finishing process is widely focused by the knowledge of forces involved in the process. This paper reports the findings about the forces acting during MAF and provides correlation between the surface finish and the forces. The resistance type force transducer (Dynamometer) has been designed and fabricated, and used to measure normal magnetic force responsible for microindentation into the workpiece and tangential cutting force producing microchips. The force data have been recorded on-line by making virtual instruments (using Lab-View software).

Key Words: MAF, Dynamometer, Force analysis, FMAB, Design of experiments

Introduction

The technology for superfinishing needs ultra clean machining of advanced engineering materials such as silicon nitride, silicon carbide, and aluminum oxide which are used in high- technology industries and are difficult to finish by conventional grinding and polishing techniques with high accuracy, and minimal surface defects, such as microcracks. Therefore, magnetic abrasive finishing (MAF) process has been recently developed for efficient and precision finishing of internal and flat surfaces. This process can produce surface finish of the order of few nanometers. The method was originally introduced in Soviet Union, with further fundamental research in various countries including Japan [1-2]. An attempt has been made to measure forces acting on the workpiece and to evolve correlation between the surface finish and forces.

In MAF, two types of forces generated by flexible magnetic abrasive brush (FMAB) are responsible for finishing: (i) normal magnetic force responsible for packing the magnetic abrasive particles and providing microindentations into the workpiece, and (ii) tangential cutting force responsible for microchipping due to rotation of the FMAB. The FMAB pushes abrasive particles downward against the workpiece surface. The relative motion between the FMAB and the workpiece is provided by rotating the magnet. As a result, the abrasive particles remove the surface material circumferentially resulting in the finished surface. The schematic diagram of the plane magnetic abrasive superfinishing apparatus is shown in Fig.1. In this process, the magnetic flux density of 0-0.44 T is used in the working gap of 1.00 –2.00 mm. Both magnetic as well tangential cutting forces are varied by changing the magnetic flux density and the working gap. The magnetic flux density is varied by changing input current to the electromagnet. On the supply of current to the magnet, the workpiece gets magnetized and magnetic lines of force emanate from the north pole of the magnet and terminate at the south pole via magnetic abrasive particles and workpiece, completing magnetic circuit (Fig.1). The space between the flat workpiece and flat-faced pole (also known as working gap/finishing gap) is filled with a mechanically made homogeneous mixture of silicon carbide abrasives (mesh no. 600) and ferromagnetic iron particles (mesh no. 300), known as unbounded magnetic abrasive particles (UMAPs) in 25:75 ratio by weight. In UMAPs, silicon carbide particles are not physically bonded to ferromagnetic iron powder. In the magnetic field, the abrasive particles can freely move around within the constraint of the adjacent ferromagnetic particles. The UMAPs are joined to each other along the magnetic lines of force and form FMAB (Fig. 2) between magnetic north pole and workpiece. This brush behaves like a multi point cutting tool. When the magnet north pole rotates, the FMAB also rotates concomitantly with the same rotational speed resulting in relative motion between the brush and the workpiece leading to finished workpiece surface. Here, cutting speed is continuously varying from the center to the periphery of FMAB.

The mechanism of superfinishing in any finishing process can be understood by the knowledge of forces involved in the process. Hence a precise force measuring device (Ring type Dynamometer) has been designed, fabricated, and calibrated as per the standard procedure [3], and used to measure forces as low as 0.5N. The force signals from the dynamometer have been acquired using a data acquisition card, and analysed by using Lab View software.

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Design of Experiments and Experimental Procedure

Experiments have been planned using statistical technique-central composite rotatable design to get useful inferences by performing minimum number of experiments. If the yield (response) 'Y_u' is a function of the levels of quantitative variables (X_{iu}), then this function can be approximated satisfactorily, within the experimental region, by a polynomial equation in the variables X_i [4]. The general form of the quadratic polynomial fitted to the experimental data is given below:

$$Y_u = b_0 + \sum_{i=1}^k b_i X_{iu} + \sum_{i=1}^k b_{ii} X_{ii}^2 + \sum_{i < j}^k b_{ij} X_{iu} X_{ju} \dots \dots \dots (1)$$

where k is no. of variables, b_i, b_{ii}, and b_{ij} are coefficients and b₀ is a constant to be evaluated. X_{iu} represents the level of the ith variable in the uth experiment. X_{ii}² is the square term and X_{iu}X_{ju} represents the interaction term.

The ranges of operating parameters have been selected from earlier studies [5] and set-up constraints, and are given in Table-1. Five independent controllable parameters thus selected are (i) current to the coil of magnet, (ii) working gap, (iii) percentage of lubricant, (iv) rotational speed of magnet, and (v) finishing time.

The workpiece fixture is mounted on the ring dynamometer (force transducer), which is clamped on the table of the milling machine and leveled by using a dial gauge (least count: 1.0 μm). The required gap between the flat-faced pole and workpiece is maintained. At the end of each experiment, the fixture and workpiece are taken out from the MAF set-up and properly cleaned by ultrasonic vibrating cleaner. The change in surface roughness value (ΔRa) is determined by measuring Ra (center line average value) before and after MAF by Taly Surf-5000 (Federal make). The difference in these two values is called ΔRa. The measurements of Ra have been done in the selected area and perpendicular to the lays obtained in the MAF process.

Response Surface Analysis (RSA)

The following response surface equations no. 2, 3, and 4 consisting of only significant parameters for magnetic force (F_m), tangential cutting force (F_c), and change in surface roughness (ΔRa) respectively have been obtained by employing MINITAB- statistical software [6].

$$F_m = 48.0 + 12.0X_1 - 7.22X_2 + 6.38X_3 - 0.58X_4 + 0.19X_5 + 3.48X_1^2 + 0.08X_2^2 + 2.51X_3^2 - 1.17X_1X_2 + 4.03X_3X_4 \dots \dots \dots (2)$$

$$F_c = 35.1 + 6.27X_1 - 6.05X_2 + 1.39X_3 - 2.45X_4 + 0.114X_5 - 1.04X_4^2 \dots \dots \dots (3)$$

$$\Delta Ra = 0.22 + 0.26X_1 - 0.041X_2 + 0.015X_4 - 0.023X_1X_2 + 0.018X_2X_4 \dots \dots \dots (4)$$

Table.1 parameters

Parameter	Unit	Range
Current (X ₁)	Amp	0.5-1.0
Working Gap (X ₂)	mm	1.0- 2.0
Lubricant (X ₃)	Wt %	1.0- 5.0
Rotational Speed of magnet(X ₄)	RPM	63-250
Finishing time (X ₅)	min	15-75

Table 2: Analysis of Variance (ANOVA) of Regression

Table.2a Magnetic Force (for Eq2)					
Source	DOF	SS	F	P	%age
Regression	20	6660.48	9.57	0	94.6
Residual Error	11	382.83			5.4
Total	31	7043.31			100
Table. 2b Tangential Cutting Force (for Eq3)					
Regression	20	2084.05	16.25	0	96.7
Residual Error	11	70.56			3.3
Total	31	2154.61			100
Table. 2c Change in Ra (for Eq3)					
Regression	20	0.106	3.93	0.012	87.73
Residual Error	11	0.015			12.27
Total	31	0.121			100

The ANOVA (Tables-2a-c) indicates that the correlation between predicted and experimental data for F_m, F_c, and ΔRa are quite adequate (87.0%-96.7%) and the calculated variance ratio (F) is more than standard F (=2.65) at 95% confidence interval (α=0.05) [7]. This variance ratio (F value) is used to measure the significance of the regression under investigation. Responses have been calculated using Eqs. (2), (3) and (4) to study the effects of various process parameters on magnetic force, cutting force, and change in Ra respectively. At the center point of the experimental design, six experiments were carried out for the same value of variables to check the repeatability of the process results. There is very little difference within the 6 values of F_m as well as F_c (Table-3). Hence, it is concluded that the process repeatability is good. Further, The experimental points (•) in Figs.3 and 4 are very close

to the computed results hence response surface analysis (RSA) results can be treated as representative experimental results.

Results and Discussion

A relationship shown in Fig.3 between magnetic force and electric current (dc) to the magnet for different working gaps has been established by using Eq.2. The trend of the curves is similar for different working gap values but the magnitude of the force is higher at lower gap. Lower gap increases the intensity of the magnetic field hence higher magnetic force. This will increase the packing density of the magnetic abrasive particles. As a result, rigidity of the FMAB increases which results in deeper microindentations into the workpiece as well as larger magnetic force. It is also observed that magnetic force increases with increase in current for a specified gap. This is so because increase in current also increases magnetic flux density in the gap resulting in increased packing density of FMAB and hence mass of abrasive particles in the FMAB gets increased. Therefore more no. of abrasive particles come in contact with the workpiece, which also contribute to the increased magnetic force.

A relationship between tangential cutting force with current for different working gaps has been established by Eq.3 and is shown in Fig. 4. The cutting force is basically a mechanical force responsible for the removal of material in the form of the microchips. This force increases with increase in current for a specified gap. This is so because increase in current increases magnetic force which increases rigidity of FMAB and depth of indentations by the particles. Therefore, resistance to the rotation of FMAB offered by the workpiece would be more and therefore cutting force increases.

Ultimately it is the force which plays most important role in any superfinishing process controlling the surface finish. Therefore in this paper the correlations between change in surface roughness (ΔRa) and both magnetic force (F_m) (Fig. 5a) and cutting force (F_c) (Fig.5b) have been established by employing STATISTICA-Software to the experimental data. The following equations have been evolved:

$$\Delta Ra = (0.0200566)(F_m)^{0.583} \dots \dots \dots (5)$$

With Correlation Factor = 0.51

$$\Delta Ra = (0.008742)(F_c)^{0.885} \dots \dots \dots (6)$$

With Correlation Factor = 0.64

It is very clear from the Fig. 5a and Fig. 5b that there is a scatter in the experimental data and hence the deviation from the calculated values. Therefore, Eqs (5) and (6) estimate approximate values. ΔRa increases with increase in magnetic and cutting force as shown in Figs. (5a) and (5b). The magnetic force is responsible for microindentation into the workpiece. The increased magnetic force enhances the strength of FMAB and indentations into the workpiece leading to increased cutting force. As a result, more will be the abrasion and more microchips will be formed resulting in better surface finish (or lower value of Ra after MAF and higher ΔRa).

Conclusions

Following inferences have been derived on the basis of above results and discussion:

Regression models for magnetic force and tangential cutting force indicate that both forces increase with increase in current. Further, ANOVA indicates that there is a good correlation between predicted and experimental data (87.73%-96.7%) (Fig. 3). It is also observed that calculated variance ratio (F) is more than the standard F value in all cases hence the proposed modes are adequate. It is also concluded that the process repeatability is good (Table.3).

Correlation between the normal magnetic force and tangential cutting force with change in Ra is comparatively weak. It needs further investigation.

Acknowledgement

The authors acknowledge the support of Department of Science and Technology, Government of India for the project on Investigations into Magnetic Abrasive Finishing (MAF) of Plane Surfaces.

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Table 3 Experiments for reproducibility

S.N.	F _m (N)	F _c (N)	ΔRa (μm)
1	46.64	34.22	0.26
2	48.58	35.82	0.21
3	43.53	34.91	0.2
4	41.59	34.22	0.21
5	50.54	34.68	0.19
6	52.40	35.13	0.28
S.D	3.640	0.66	0.027

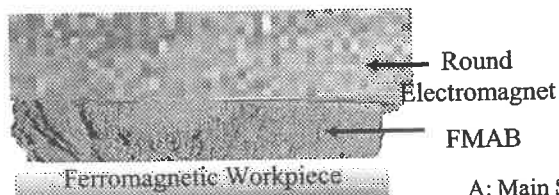


Fig.2 Digital Photograph of Flexible Magnetic Abrasive Brush (FMAB)

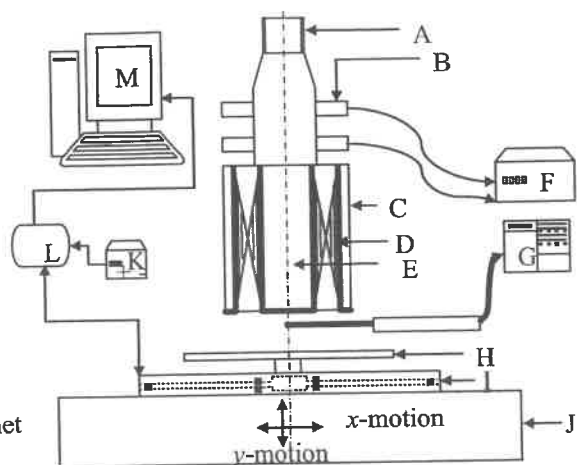


Fig.1 Schematic View of Plane Magnetic Abrasive

A: Main Spindle of Vertical Milling Machine; B: Slip Rings; C: Outer Pole (South Pole); D: Coil; E: Central Pole (North Pole); F: DC Power Supply; G: Digital Gauss Meter; H: Workpiece Fixture; I: Ring Dynamometer; J: Table of the Machine; K: 6.0 Volt DC Supply for Signal Conditioning Unit; L: Signal Conditioning Unit; M: PC for recording force data

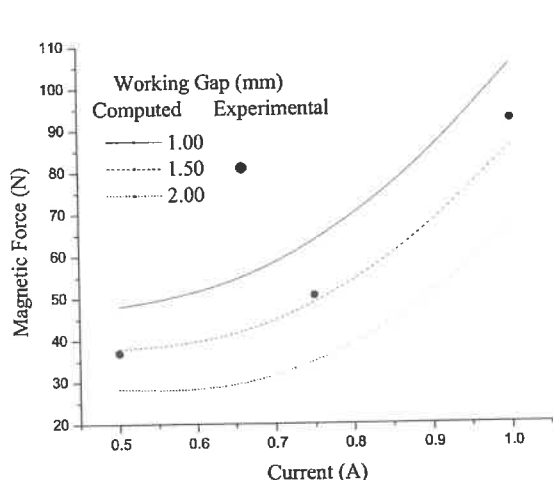


Fig.3 Effect of current on magnetic force for different gap at %oil = 3, RPM = 125, Time = 45min

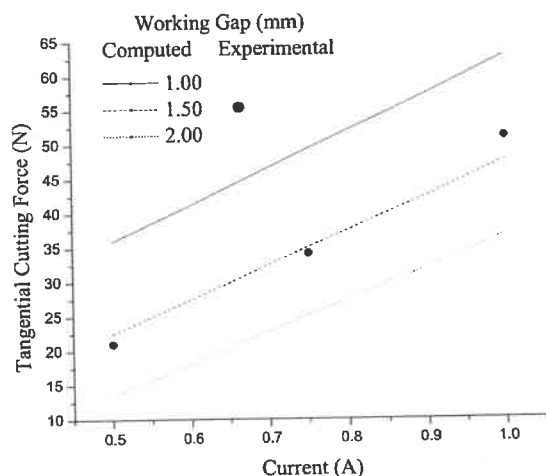


Fig.4 Effect of current on Tangential cutting force for different gap at %oil = 3, RPM = 125, Time = 45 min

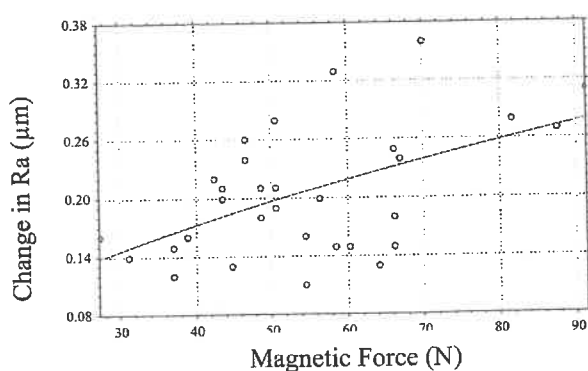


Fig.5a Correlation between Magnetic Force and Change in Ra

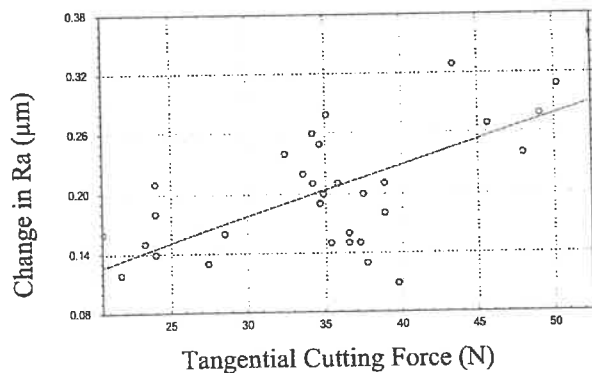


Fig.5b Correlation between Tangential Cutting Force and Change in Ra

INTERNAL MAGNETIC ABRASIVE FINISHING FOR AUSTENITIC STAINLESS STEEL TUBES BENT USING HIGH FREQUENCY INDUCTION BENDING

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Abstract

This paper examines the applicability of magnetic abrasive finishing to the internal finishing of large-sized bent tubes produced by high frequency induction bending. Finishing equipment was developed for large-sized bent tubes. The finishing experiments using austenitic stainless steel tubes bent by both cold-drawing and high frequency induction bending demonstrate the finishing characteristics and reveal the differences in the finishing mechanism between the two types of elbows. The elbow made by cold-drawing required only one finishing process to achieve a $0.03 \mu\text{m } Ra$ surface. In contrast, high frequency bent elbow, which is covered with an oxide film, required two-stage finishing using two different sizes of ferrous particles for a smoothly finished surface, $0.05\sim 0.13 \mu\text{m } Ra$.

Keywords: Internal finishing, Magnetic abrasive finishing, Austenitic stainless steel bent tube, Finishing characteristics, High frequency induction bending, Two stage finishing

1. Introduction

High frequency induction bending is a hot bending process for pipes and shaped steels using high frequency induction heating. The process has the advantage of three-dimensional flexibility in bend radius and angle within the limits of the bending machine. Complex-shaped tubes, which consist of both bent and straight sections, produced by high frequency induction bending, are used for piping systems to minimize the number of welded joints, thereby reducing the production and installation costs and improving the quality of the piping system. The tubes are required to have smoothly finished inner surfaces to prevent the contamination of gas and liquid. However, the complexity of the tube shapes and the oxide film generated on the tube surface during high frequency bending considerably increase the difficulty of internal finishing of the tubes with conventional technologies.

An internal magnetic abrasive finishing process was proposed for producing precisely finished inner surfaces of nonferromagnetic tubes used for piping systems [1]. This paper examines the applicability of the magnetic abrasive finishing to the internal finishing of large-sized bent tubes produced by high frequency induction bending. Finishing equipment was developed for large-sized bent tubes, and the performance of the equipment was examined through finishing experiments using SUS316 and SUS304 austenitic stainless steel elbows made by both cold-drawing and high frequency induction bending. They demonstrate the finishing characteristics, including surface roughness and material removal, and reveal the differences in the finishing mechanism between the two types of elbows.

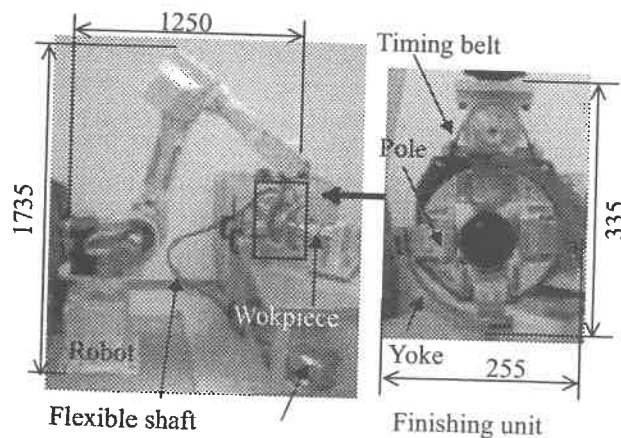


Fig. 1 External photograph of experimental setup

2. Processing principle and experimental setup

2. Processing principle and experimental setup

The poles, which consist of small permanent magnets, placed outside the bent tube generate the magnetic field needed for attracting the magnetic abrasive to the finishing area by magnetic force. When the poles rotate around the bent tube, the magnetic abrasive, driven by magnetic force, rotates along the inner surface of the bent tube along with the poles, and removes material from the surface. Manipulating the rotating poles along the tube axis causes the magnetic abrasive to follow the poles' motion, finishing the entire inner surface of the tube [2].

Table 1 Experimental conditions

Workpiece	SUS316 stainless steel 90° elbow OD: 76.3 mm, ID: 72.3 mm, Radius of curvature: 400 mm
Ferrous particles	Electrolytic iron particles: 56 g (330 μm in mean dia.)

Previous research developed the finishing equipment and demonstrated the process applicability to nearly uniform internal finishing of SUS304 stainless steel tubes bent by a cold-drawing process (16~21 mm outer diameter, 30~80 mm radius of curvature) [2]. Finishing equipment for large-sized bent tubes, shown in Figure 1, was developed based on the previous equipment. The robot has 6 axes of motion to accommodate tubes of various configurations, and the finishing unit is connected to the arm of the robot.

3. Finishing Characteristics

3.1 Cold-drawing processed elbows

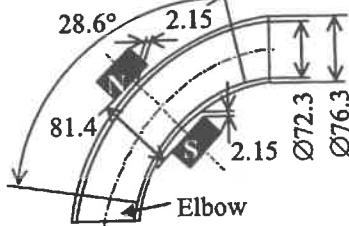
Before the internal finishing of the tube processed by high frequency induction bending, the performance of the equipment was examined through the internal finishing of cold-drawn elbows. Table 1 shows the experimental conditions. A mixture of ferrous particles and WA magnetic abrasive was introduced with lubricant into the elbow at the beginning, and lubricant was supplied into the mixture at every stroke of the finishing unit. It took 5 min for one stroke. The magnetic field measured at the finishing area is shown in Fig. 2, the intensity of which was more than 20% greater than in the case of the previous equipment.

Figure 3 shows the changes in surface roughness at outside, inside, lateral (upper), and lateral (lower) regions and material removal with finishing time. Both the lateral regions show almost same value. The surface was uniformly finished; which is also shown in Fig. 4. This demonstrated both the process applicability to the large-sized elbow and the efficiency of the equipment.

3.2 High frequency induction processed straight tubes

The process applicability to bent tubes made by hot bending and specifically large-sized tubes remained unknown to this point. To reveal the fundamental finishing characteristics, finishing experiments were initiated with straight tubes

Table 1 Experimental conditions

Workpiece	SUS316 stainless steel 90° elbow OD: 76.3 mm, ID: 72.3 mm, Radius of curvature: 400 mm
Ferrous particles	Electrolytic iron particles: 56 g (330 μm in mean dia.)
WA magnetic abrasive	14 g (80 μm in mean dia.)
Pole	Nd-Fe-B rare earth permanent magnet: $49 \times 35 \times 26$ mm
Pole rotation	400 min^{-1}
Pole feed	0.5 mm/s
Lubricant	Soluble type barrel finishing compound: 9.5 mL / stroke
Finished area	

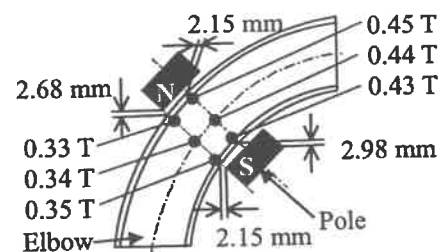


Fig. 2 Magnetic flux density at finishing area

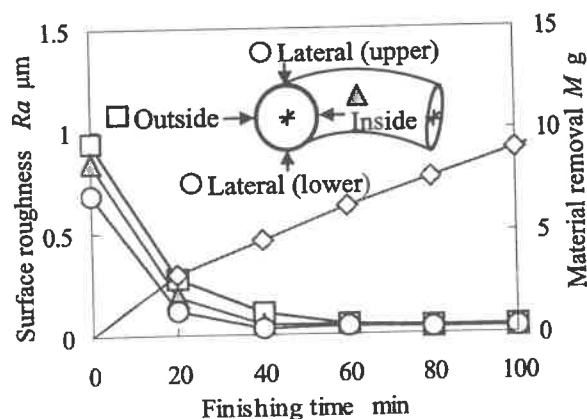


Fig. 3 Changes in surface roughness and material removal with finishing time

processed by high frequency induction processing ($\varnothing 89.1 \times \varnothing 83.1 \times 400$ mm). Figure 5 shows SEM microscopy and oxygen distribution by EPMA of the cross section of the tube. The oxide film, about $3 \mu\text{m}$ thick, is obvious on the inner surface, and needs to be removed in order to obtain smooth surface.

Figures 6 and 7 show the experimental results with 330, 510, and 1680 μm in mean particle diameter. The conditions with 330 μm iron particles are the same as those for the cold-drawn tube, shown in Fig. 3; this produced the least material removal of the three conditions. According to Fig. 7 (a), the relatively longer wavelength components of the roughness profiles remain on the surface after the finishing process. This resulted from the accumulation of the small cutting operations sustained by a small finishing force. The magnetic force acting on the ferrous particles, controlling the finishing force of the magnetic abrasive, is a function of the particle volume. This resulted in greater material removal in the cases of 510 and 1680 μm iron particles than in the case of 330 μm iron particles. Deep cutting marks resulted from the greater finishing force are observed on the finished surfaces in Fig. 7 (b) and (c). However, these conditions were inappropriate to obtain a smoothly finished surface. Accordingly, two stage finishing was proposed to the tube finishing with a combination of larger iron particles for coarse finishing in the first stage and smaller iron particles for fine finishing in the second stage.

Figure 8 shows the finishing characteristics by two stage finishing with 1680 μm iron particles for the first 40 min and 330 μm for the final 80 min. While the first stage shows similar characteristics to those in Fig. 6, the material removal dropped in the second stage and resulted in the fine surface finish, 0.04 μm R_a . This demonstrates the feasibility of two stage finishing for internal finishing of elbows made by high frequency bending and covered with an oxide film.

3.3 High frequency induction processed elbows

Figure 9 shows the changes in surface roughness and material removal with finishing time for an elbow finished by the two stage finishing with 1680 and 330 μm iron particles.

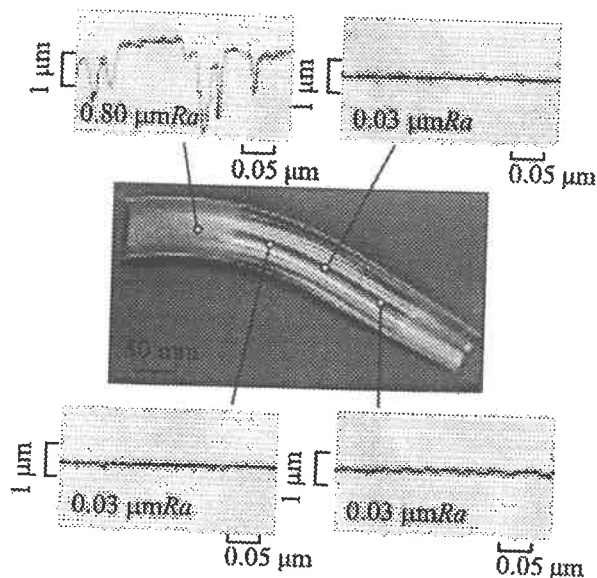


Fig. 4 Photo of inside the SUS316 stainless steel bent tube after finishing

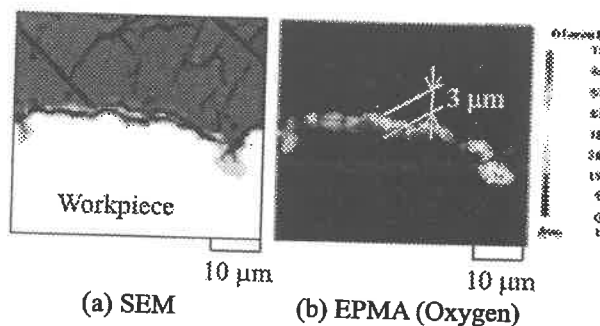


Fig. 5 SEM microscopy and EPMA of cross section of SUS304 stainless steel tube

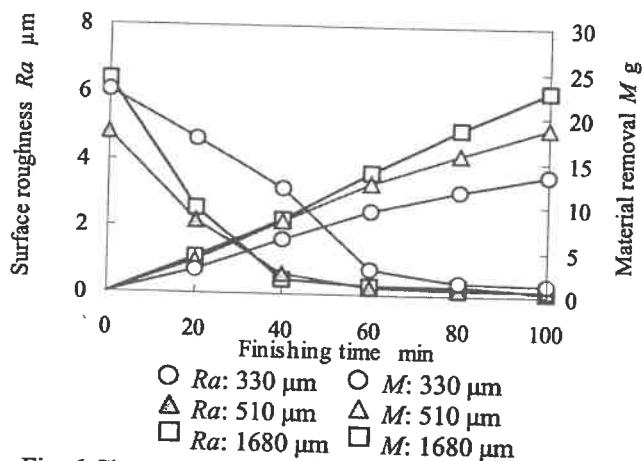


Fig. 6 Changes in surface roughness and material removal with finishing time

[Conditions] Workpiece: SUS304 stainless steel tube: $\varnothing 89.1 \times \varnothing 83.1 \times 400$ mm, Electrolytic iron particles (330, 510, 1680 μm in mean dia.): 56 g, Pole-pole distance: 93.4 mm, other conditions: see Table 1.

The unevenness of the finished surface is seen in the first stage, outside: 6.49 to 4.70 μm R_a , inside: 10.2 to 6.06 μm R_a , lateral (upper): 8.22 to 3.74 μm R_a , and lateral (lower): 9.21 to 4.42 μm R_a . After the second stage, the surface regions were finished to 0.13, 0.11, 0.05, and 0.05 μm R_a , respectively. Although there remains some surface quality unevenness, this study proves the process applicability to internal finishing for not only fine finishing but also coarse finishing for a wide variety of complex-shaped tubes made by high frequency induction bending.

4. Conclusions

While obtaining uniformly smooth finished surfaces in cold-drawn elbows, the same finishing conditions scarcely removed the oxide film generated by high frequency induction bending. This resulted in the necessity for different finishing conditions to remove the work-hardened layer including an oxide film and primary material. Greater magnetic force acting on the magnetic abrasive was required to remove the work-hardened layer than that for merely removing primary material, and this force was controlled by the size of the ferrous particles mixed with the magnetic abrasive. Accordingly, two stage finishing using two sizes of ferrous particles achieved the fine finish for elbows made by high frequency bending.

Acknowledgement

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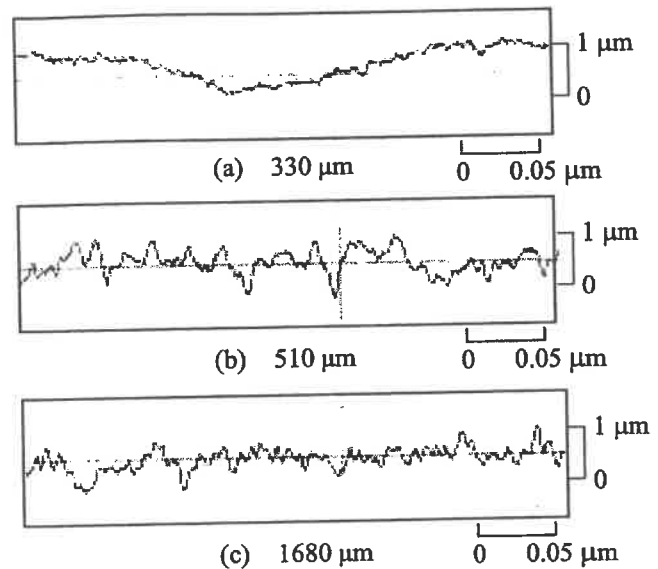


Fig. 7 Surface roughness profiles after finishing for 100 min

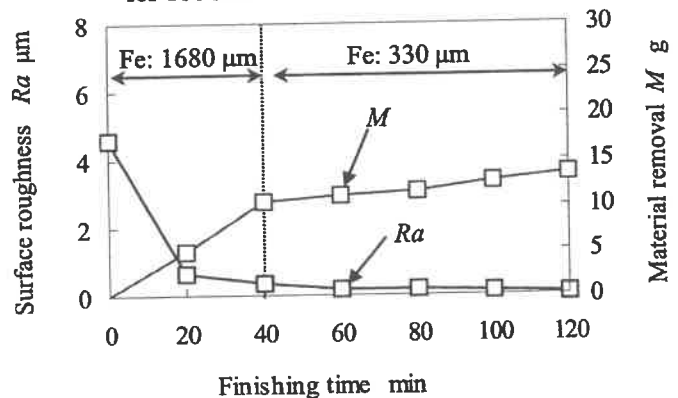


Fig. 8 Changes in surface roughness and material removal with finishing time in two stage finishing

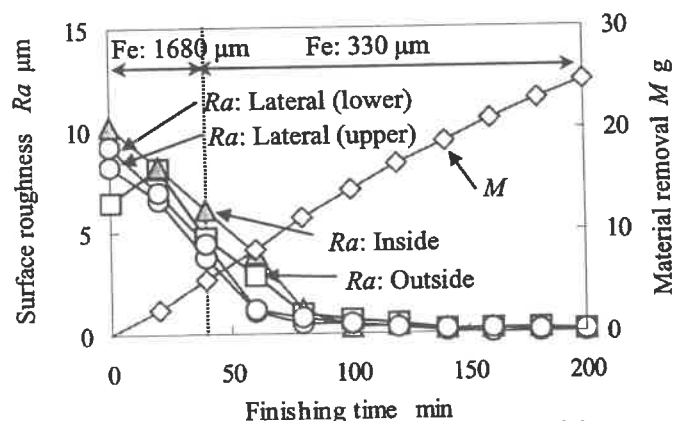


Fig. 9 Changes in surface roughness and material removal with finishing time

[Conditions] Workpiece: SUS304 stainless steel 90° elbow (89.1 mm OD, 83.1 mm ID, 400 mm Radius of curvature), Electrolytic iron particles: 1680, 330 μm in mean dia.): 56 g, Pole-pole distance: 94.7 mm, other conditions: see Table 1.

LAPPING OF FLAT SURFACES WITH TWO-METALLIC TOOLS

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1. Introduction

Lapping is one of the most common finishing operation used on flat or cylindrical surfaces. Fine surface finish and high dimensional accuracy can be obtained in lapping with the use of relatively simple means of productions. Geometrical structure of the lapped surface has good properties in sliding joints - the valleys in the surface roughness of the contacting bodies can serve as local reservoirs for lubricants, as well as in stationary joints because of the high bearing ratio. Elements of this machining system are characterised by the set of many structural, material and surface properties connected with the workpiece, lap, abrasive particles and the lapping machine. The lap - a tool used in lapping - has an essential influence on the dimensional and shape accuracy [6,7,13].

Lapping and polishing using loose abrasive are classified as the basic finishing operations. These technologies can be used on metallic parts, as well as on non-metallic machine elements, mainly on engineering ceramics which have found use in many engineering applications. There is no limitation on material which can be lapped or polished [1,4,5]. Metallographic sections belong to the wide range of their use. Multi-directional lay, low temperature of the process, relatively low unit pressure and lapping speed help to obtain high dimensional accuracy and fine surface finish. The quality and efficiency of lapping depends on many factors connected with the lap, lapping machine and test conditions. Free charging with abrasive [8] (the abrasive particles become embedded in the lap during lapping - conventional lapping), as well as machine forced charging before lapping [9] can be done on an active surface of the tool.

2. Lapping with tools of cast iron freely charged with abrasive

This method of machining, apart from many advantages, has an essential disadvantage related to the abrasive contamination mainly of the soft materials or soft structural constituent. The abrasive contamination of the surface layer is damaging during the operating period and causes increasing abrasive wear of elements in sliding joints [2,3]. When the abrasive compound is dosed periodically, the material removal rate decreases quite rapidly. Introducing abrasive grains of different numbers (size) intensifies the material removal but it also affects the quality of the lapped surface.

The probability of breaking the grains up increases with the increase in the micrograin size, under the constant load and abrasive particles concentration. In order to ensure the one-layer distribution of the large number micrograins it is necessary to reduce the abrasive particles concentration on the active surface of the lap and in turn abrasive particles crushed under the higher unit pressure and the material removal rate decreases. When the abrasive is supplied from the paste of abrasive compound periodically, the removal rate and the surface roughness decrease due to the crushing of micrograins and the charging of the active surface of the lap with abrasive stabilises in the final phase of lapping.

3. Lapping with two-metallic tools

Good abrasive properties of two-metallic laps could be explained by the fact that the micrograins supplied on an active surface get an adequate angular acceleration what makes the machining more intensive, then they penetrate the tool surface and eventually become embedded in the softer material. Material can be removed by particles embedded in the lapping plate as well as by rolling grains [10]. Two-metallic tools of cast iron-copper and cast iron-steel were used in lapping experiments and a monometallic lap of cast iron was used for comparison experiments. The main frame of tools made of Z_s50007 cast iron (183 HB) and metallic inserts of 45 steel (387 HB) and MOOB copper (80 HB) were joined together with glue. Surface area of metallic inserts took 50% of an active surface of the tool (Fig.1 and 2). The highest removal rate was observed for cast iron-steel lap and the lowest for monometallic lap. The results of testing on two-metallic tools show that material removal increases considerably with the increase in unit pressure and lapping speed (Fig.3). This is the results of deeper penetration of the workpiece surface by abrasive particles and the results of the change in the dynamic interaction of abrasive. Also the higher the relative speed in the lap-workpiece system is, in the period of time, the longer the cutting trajectories are. The higher material removal rate was also recorded during experiments performed on samples of NC6 steel using cast iron-steel lap. The same is true for experiments performed on samples of Z_s55003 cast iron, however the use of cast iron-copper lap for relatively hard samples of NC6 steel is not as effective. The lowest surface roughness was obtained for cast iron-copper tool and the highest roughness for cast iron-steel lap.

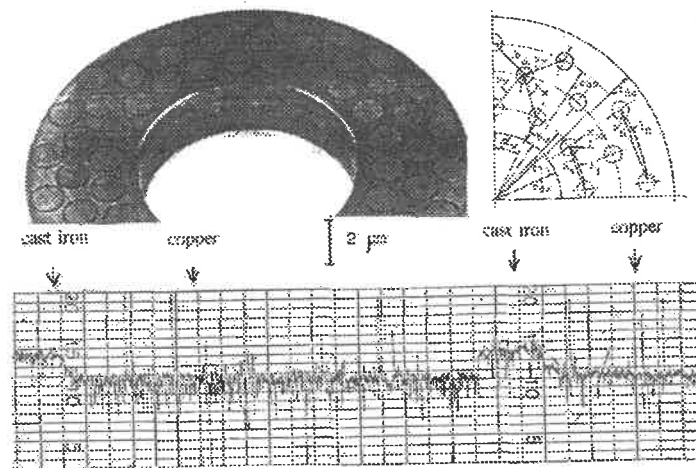


Fig.1. Construction of two-metallic lapping tools

4. Lapping with abrasive-metallic tools

Abrasive particles are introduced into the contact region between the workpiece and the lapping tool drop by drop or periodically when monometallic and two-metallic laps are used. Some of micrograins are removed from the active surface of the lap by moving workpieces and separators and they do not take part in machining. Particles are usually dosed excessively in practice. Taking into accounts conditions of conventional lapping, promising approach to lapping is to utilise non-conventional constructional solutions of lapping tools [11,12].

An abrasive-metallic lapping plate is one of the examples. Only cutting fluid (based on machine oil and kerosene) is dosed on the tool during lapping. The main frame of the tool is

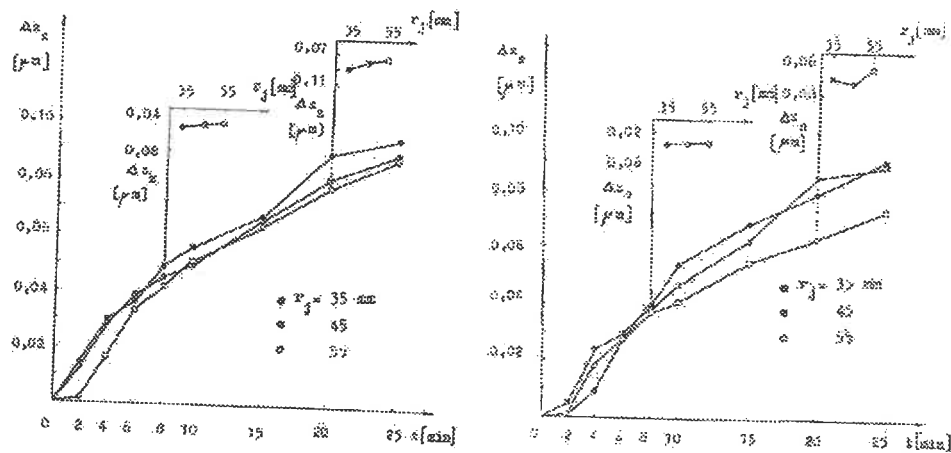


Fig.2. Wear (for r_j diameter) of two-metallic lapping tools (z_z – Zs50007 cast iron, z_s – 45 steel) at machining of cast iron Zs55003 (99C F400/17, $p=0.23$ MPa., $v=0.15$ m/s)

made of ZI250 cast iron (191 HB) with ferritic-perlite matrix. In the first version of the tool, abrasive inserts consists of silicon carbide or aloxite grits and are located on the active surface symmetrically. Their contours can be rectangular or circular. In the second version, circular abrasive inserts consist of BC 400 micrograins mixed with electrographite or copper micrograins and epoxy resins. The effect of using abrasive-metallic laps based on the 99C F320/29 abrasive (hardness J) is shown in Fig.4. The material removal rate is twice as high as in conventional lapping but the quality of lapping decreases – surface roughness is lower. Designed abrasive-metallic tools are characterised by the possibility of using cutting ability of particles from the worn inserts. The active surface of the cast iron frame can be charged with these micrograins. The proper selection of inserts characteristic can make easier the optimisation of lapping what is very important especially when superhard grits are used due to the high cost.

5. Conclusion

The results of testing enable formulation of the following conclusions:

- higher material removal was recorded for conventional lapping but machine forced charging of laps with abrasive makes possible obtaining the surface roughness lower
- using two-metallic laps, the highest removal rate was observed for cast iron-steel tool, and the lowest surface roughness was obtained for cast iron-cooper lap,
- the high rate of material removal was achieved when metallic tools with abrasive inserts were used but the quality of lapping decreased as compared to surface roughness obtained in conventional lapping.

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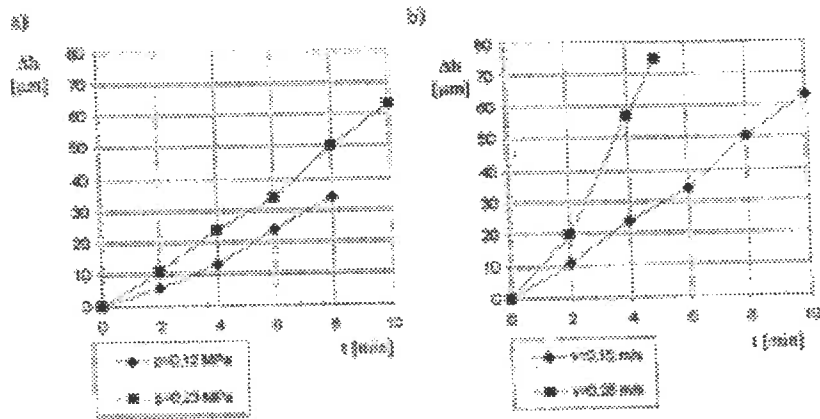


Fig.3. Material removal rate as a function of time for lapping of: a) cast iron Zs55003, $v=0.15$ m/s, b) steel NC6, $p=0.23$ MPa; test conditions: forced charging with abrasive of 99C F400/17, charging conditions: $p=777$ N/m, $t=180$ s, $v=0.059$ m/s

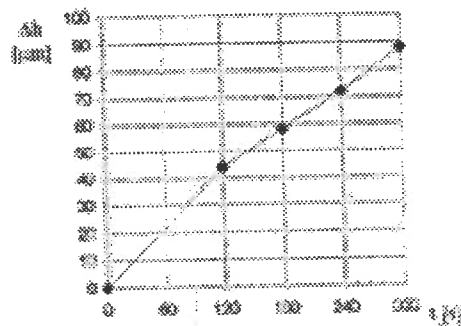


Fig.4. Material removal rate in lapping of NC6 steel using abrasive-metallic tools; test conditions: $v=1$ m/s, $p=0.169$ MPa

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Chemical-Mechanical Planarization of an SU-8/Copper Combination for MEMS

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Key Words: CMP, planarization, polymer, copper

Abstract: Chemical-mechanical polishing (CMP) is a technology for wafer level planarization. Typically, CMP is applied to a wafer with two different materials at its surface, balancing the difference in material properties by an appropriate mix of mechanical machining and chemical attack. This paper investigates the application of CMP for planarizing electroplated copper (Cu) embedded in a photosensitive epoxy material, which can be patterned by means of photolithography (SU-8). Such a material combination is very advantageous for fabricating electromagnetic Micro Electro-mechanical Systems (MEMS). The goal was to achieve an optimal local planarization, i.e. a minimal step height between the two materials within each structure. Some of the key parameters influencing it were optimized. The parameters investigated to achieve an optimal local planarization were the pad hardness, the baking temperature ("hard bake") of the SU-8 which affects the material's hardness, and the density of the structures.

1. Introduction

Chemical-mechanical polishing (CMP) allows a planarization of wafers with two or more materials of different hardness at its surface [1]. Subject to the investigation was to apply CMP for planarizing electroplated copper (Cu) embedded in epoxy material. This material combination is widely used in the fabrication of Micro Electro-mechanical Systems (MEMS) [2]. In this particular case, Cu was used to form double layer coils and vias for a linear microactuator. It was electroplated in micromolds of photosensitive epoxy (SU-8) [3, 4]. This polymer material not only is well suited for photolithographic patterning, it also features excellent mechanical properties including machinability after being subjected to a baking process [5].

After electroplating, metal structures usually feature an uneven surface. Furthermore, electroplating typically has a non-uniform deposition rate over the wafer surface. As a result, some micromolds are filled faster than others leading to thickness variations on the wafer. To achieve a constant film thickness, the Cu was allowed to "overgrow" the micromold (Fig. 1). However, such an approach requires a polishing step to remove excessive metal and to achieve a flat surface [6]. The process of choice to do so can be derived from a Metal-CMP process well established in the semiconductor industry, which was developed to planarize Cu or other metals embedded in interlayer dielectrics [7].

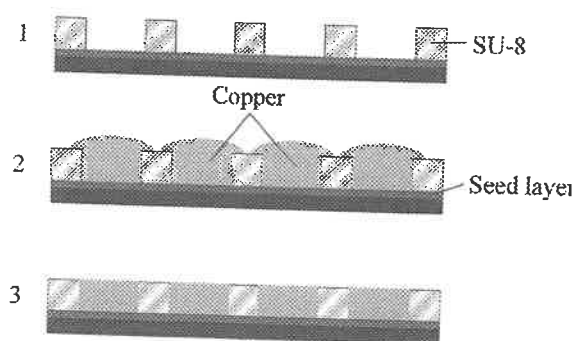


Figure 1: Approach for the planarization of a SU-8/Cu combination: 1: SU-8 patterning, 2: Cu electroplating, 3: CMP

The purpose of CMP is to achieve planarization, separated into local planarization creating plane surfaces for each component and global planarization aiming at planarizing the whole wafer. The main goal of this work was to achieve local planarization.

The parameters affecting a CMP process may be separated into tool, process, and workpiece related ones. In the area of tools, the slurry and pad composition are the most important ones. While a standard slurry is available for machining Cu, a process optimization is mainly accomplished by selecting an appropriate pad, which is one of our optimization goals. For choosing the process parameters like rotational velocity and pressure, it may be useful to derive them from similar applications. In our case, they were based on prior work on planarizing permalloy embedded in SU-8 [8]. In the area of the workpiece, both mechanical properties like hardnesses and Young's moduli of the materials involved are of importance, as is the spacing of the metal structures in the embedding material. The mechanical properties of electroplated Cu practically don't vary. However, the hardness of the SU-8 may be influenced by the temperature and the exposure time of a thermal treatment necessary to bake out the material after its development. Defining an

optimal postbake temperature was the second goal of these investigations. Furthermore, to account for the structure spacing, the tests were performed with different test structures.

2. Experimental Setup and Procedure

To investigate the CMP of the Cu/SU-8 combination, a Peter Wolters 3R40 polishing machine adapted for CMP was used. The plate diameter was 400 mm, the load was applied through weights. The rotational speed of both plate and wafer was 18 min^{-1} . In all experiments, the Metal-CMP slurry (Rodel MSW1500) was used at a flow rate of 15 ml/min. The test silicon wafers had a diameter of 4". For investigating the pad composition, three pads varying in hardness were used. For processing the SU-8, a range of postbake temperatures were applied, resulting in a variation of the hardness of the SU-8. A difference in the density of the structures was achieved by using two types of test structures. The first one is a coil with two different spacing widths (5 μm and 60 μm) (Fig. 2a) and the second one was a quad arrangement of vias, i.e. through connections (approx. 30 μm x 40 μm footprint per via, 410 μm between vias and 1530 μm between groups of vias (Fig. 2b)).

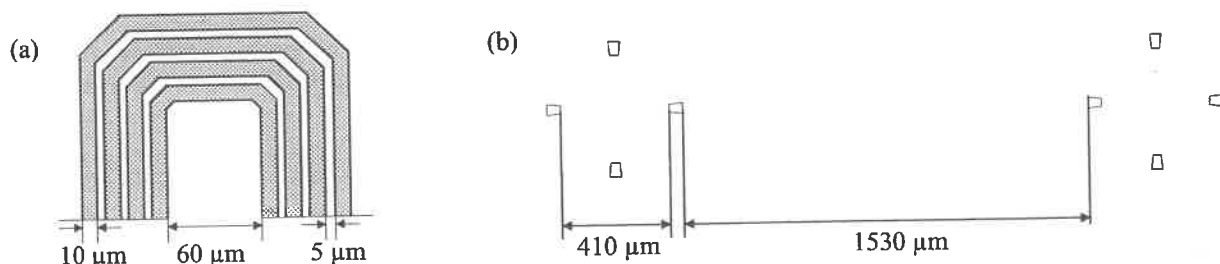


Figure 2: Top view of the two test structures. a: structure type 1 (coil), b: structure type 2 (vias). Please note the different scales

3. Influence of the Pad Hardness

For these experiments, standard pads of different hardnesses (Rodel Politex® Supreme, Suba 1200, and IC 1400) were used. Tab. 1 depicts the physical properties of the pads, with their hardness and compressibility being most influential on the local planarization results., i.e. the step height between Cu and SU-8. The pad hardness investigations were conducted for structure type 1 and 2. Furthermore, the hardbake conditions were not varied, the SU-8 hard bake temperature was 120°C for structure type 1 and 150°C for structure type 2, respectively.

TABLE 1
PROPERTIES OF POLISHING PADS

Pad	Politex	Suba 1200	IC 1400
Material	Polyurethane	Polyurethane impregnated felt	Polyurethane foam, urethane foam
Composition	Pore structure on compressible substrate	Single layered	Double layered
Hardness [Shore A, D]	Not specified	80 (A)	57 (D)
Compressibility [%]	Not specified	6	2.25

The softest pad used during the investigations was Rodel Politex® Supreme. It is softer than the SUBA and IC-pads, (also offered by Rodel) due to its material and porous structure and was originally developed for final silicon wafer polishing. No hardness or compressibility figures are available for this pad. Figure 2 (left) depicts the results of polishing. The step height was in the order of 1,200 nm if the space between the structures was 60 μm and 250 nm if it was 5 μm . The surfaces of the Cu structures were convex. The surface of the SU-8 has a curved shape with a smooth transition to the Cu structures.

The medium pad Suba 1200 was originally developed for stock and intermediate polishing during silicon wafer fabrication. Its material is polyurethane. For the medium pad (fig. 2, left) the step heights were 200 nm (60 μm distance) and 90 nm (5 μm distance), respectively. Compared to the results with the softer pad, this pad creates a higher roughness. Furthermore, the Cu structures feature a sharper edge and a step transition between the SU-8 and the Cu surfaces.

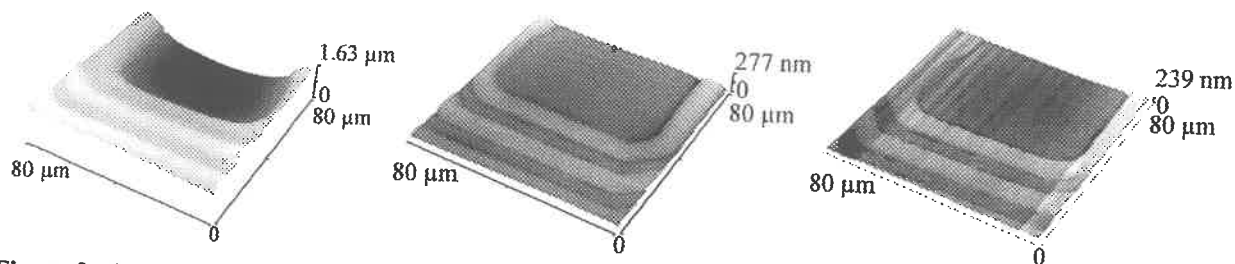


Figure 3: AFM topography analysis of the SU-8/Cu combination after machining with a Politex® Supreme pad (left), after machining with a Suba 1200 pad (middle), after machining with a IC 1400 pad (right)

The hardest pad was IC 1400, explicitly developed for the planarization during semiconductor fabrication. It is a two layer design with a harder upper layer made out of polyurethane on a softer lower layer made of urethane foam. The hardness of the polyurethane upper pad is higher than the one of the Suba 1200. The step height was in the range of app. 100 nm if the space between the structures was 60 μm and 55 nm if it was 5 μm.

Table 2 summarizes the results of the AFM investigations for the three pads and for the two pattern distances of structure 1 (5 μm and 60 μm distance between leads) and the two pattern distances of structure 2 (410 μm and 1530 μm distance between groups of vias). While harder pads yield better results in respect to the local planarization, they cause higher roughness due to their more aggressive mechanical interaction between pad, abrasives, and workpiece. Although a better planarization has to be achieved in the future to meet the requirements for photolithography with a step height lower than 20 nm, this investigations show clearly the relationship between pad hardness, resulting surface roughness, and achievable planarization. Also, the distance between structures influences the step height between structure and embedding material. Therefore, it is of advantage to develop design rules for structure size and spacing between structures to further optimize local planarization.

TABLE 2
STEP HEIGHT AND ROUGHNESS RESULTS FOR DIFFERENT POLISHING PADS

Pad	Politex	Suba 1200	IC 1400
Structure type 1			
Step height (5 μm structure distance) [nm]	app. 250	app. 90	app. 55
Step height (60 μm structure distance) [nm]	app. 1,200	app. 200	app. 100
Cu roughness R_a [nm]	2.1	3.1	3.8
SU-8 roughness R_a [nm]	1.3	2.3	2.4
Structure type 2			
Step height (410 μm structure distance) [nm]	app. 455	app. 200	app. 150
Step height (1530 μm structure distance) [nm]	app. 1,050	app. 285	app. 195
Cu roughness R_a [nm]	2.1	3.1	3.8
SU-8 roughness R_a [nm]	1.1	1.5	1.6

4. Influence of the SU-8 Hardbake Temperature

The basic idea behind influencing the SU-8 mechanical properties through its hardbake temperature is to harmonize the removal rates of both materials in order to optimize the local planarization. To compare the removal rate results and the achievable degree of planarization as a function of the baking temperature, CMP tests were conducted. Wafers containing structures 2, with the SU-8 baked at 95°C, 120°C and 150°C were polished using the hardest of the pads (IC 1400), which according to the previous tests achieved the best local planarization results. Figure 4 shows AFM analyses of the topography after machining.

Corresponding to the distance dependant step height measurements in chapter 3, in this application the step height between SU-8 and Cu for two pads in a group (distance 450 μm) and for two neighboring groups (distance 1570 μm) was measured. Table 3 summarizes the results of the AFM investigations for the three hard bake temperatures

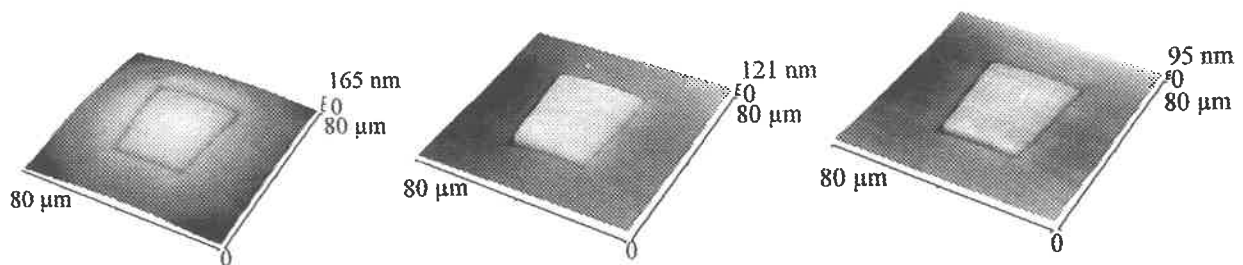


Figure 4: AFM topography analysis of the SU-8/Cu as the SU-8 was treated with different temperatures at hard bake process step: 95°C (left), 120°C (middle), 150°C (right)

Applying a 150°C hard bake led to the best planarization result with a step height of 150 μm for two neighboring vias with a distance of 410 μm. As also observed during the pad hardness investigations the step height increased, when the distance between two protruding structures increased.

TABLE 3
STEP HEIGHT AND ROUGHNESS RESULTS FOR STRUCTURE 2 AND
DIFFERENT HARD BAKE TEMPERATURES

Hard bake temperature	95°C	120°C	150°C
Step height (410 μm structure distance) [nm]	app. 240	app. 185	app. 150
Step height (1530 μm structure distance) [nm]	app. 380	app. 310	app. 195
SU-8 roughness R_a [nm]	2,5	2,2	1,6

5. Conclusion

The investigations showed that CMP is suitable to planarize a Cu/SU-8 combination. It was found that the hardest pad achieved the best local planarization (app. 50 nm-100 nm). The price to be paid is a greater roughness R_a (app. 4 nm for Cu and 2.5 nm for SU-8), roughly double the value achieved by the softest pad. The step height was smaller for greater density of structures (or smaller distances), and accounted for an optimum of 55 nm for two neighboring structures with a 5 μm spacing filled with SU-8. A second approach to optimize the planarization is to exploit the hardening effect of a SU-8 thermal treatment. With a higher hard bake temperature, step height and roughness decreased. The best result (a step height of 150 μm) for two structures with a 410 μm spacing was achieved using a hard bake temperature of 150°C. These results indicate, that chemical-mechanical polishing of Cu embedded in SU-8 can be executed with acceptable results regarding the step height.

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POLISHING PROCESSES FOR STRUCTURED SURFACES

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Abstract

(Chemo-) mechanical polishing, abrasive flow machining and laser polishing are applied to finish structured molds for injection molding of optical elements. The paper focusses on the development of a (chemo-) mechanical polishing process for structured surfaces that requires specially shaped tools and process strategies which differ from conventional polishing of not structured surfaces. Machining results, i. e. surface roughness and form accuracy of the polished structures, are compared to that achieved by abrasive flow machining and laser polishing.

Introduction

Precision molds for the replication of structured optical elements are generated by precision machining processes like diamond machining or precision grinding. To achieve highest surface quality the application of polishing processes to improve surfaces roughness of the structured surfaces can be necessary [1]. Within this work, possible polishing processes as subsequent finishing operations are evaluated and (chemo-) mechanical polishing, abrasive flow machining as well as laser polishing are applied to structured surfaces.

Structured steel (X40 Cr13) and electroless nickel plated steel samples as shown in fig. 1 were used for all polishing experiments. The workpieces exhibit linear V-grooves with depths d and widths w between 0.1 mm and 1.5 mm and included angles α of 60°, 90° and 150°. The samples were premachined by grinding and diamond fly-cutting, resp. White-light interferometry (WLI) was used for measuring surface roughness and form accuracy of the structures.

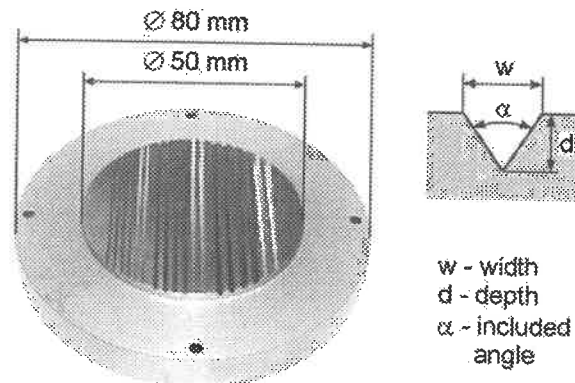


Fig. 1: Sample with linear V-grooves.

(Chemo-) mechanical polishing of structured surfaces

(Chemo-) mechanical polishing of structured surfaces requires dedicated tools and process strategies which are different from conventional polishing of continuous, i. e. not structured surfaces. At the Laboratory for Precision Machining LFM a (chemo-) mechanical polishing process for finishing of rectangular and V-grooves using pin type and wheel type polishing tools is in development [2]. The polishing tools are made of the polyamid PA6 that incorporates relatively high hardness to guarantee form stability and sufficient toughness so that the abrasives can be impressed into the tool. Furthermore, PA6 can be machined very precisely by means of diamond turning. For performing the polishing experiments a Precitech Microfinish 300 aspheric polishing machine was

modified, now consisting of three linear axes as well as polishing and a workpiece spindle. For the application of both pin type tools and polishing wheels the polishing spindle can be tilted by $\pm 90^\circ$.

Conical pin type tools with an included angle (2κ) of 140° and conical wheel type tools with a conic angle (2κ) of 30° were used for polishing of one side of 0.2 mm deep 150° -grooves (fig. 2). To achieve full contact with the one side of the

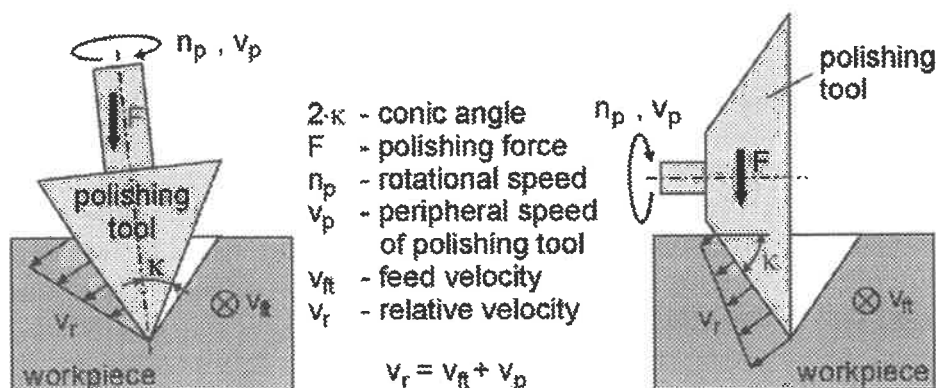


Fig. 2: Polishing tools for structured surfaces.

structure the conical pins have to be tilted by 5° . 2 mm long groove sections were polished with varying process parameters. The polishing time was varied between 1 min and 15 min, polishing forces F between 0.02 N and 0.3 N and the rotational speeds n_p of the polishing spindle between 100 rpm and 1000 rpm (in case of conical pin type tools) or 25 rpm and 100 rpm (in case of polishing wheels), i. e. relative velocities v_r between 0.01 m/s and 0.1 m/s for polishing both the steel and electroless nickel plated substrates. A water based diamond suspension with a grain size of $0.1 \mu\text{m}$ was used.

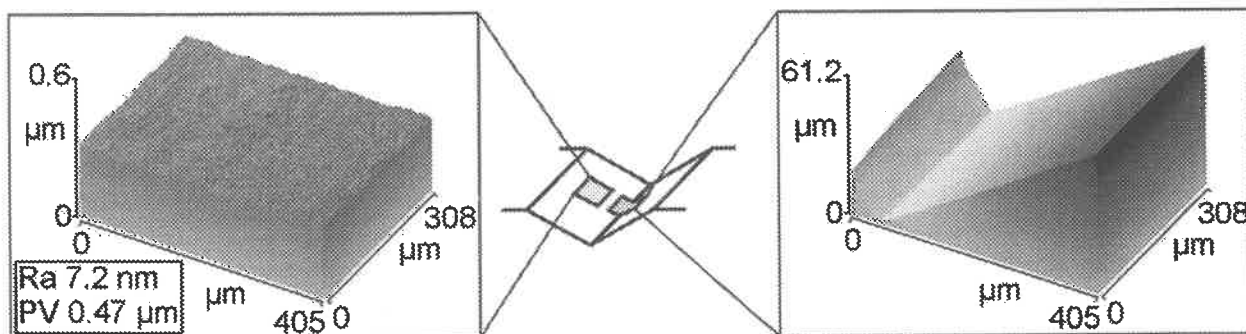


Fig. 3: (Chemo-) mechanical polished side of a V-groove in electroless nickel.

Fig. 3 shows a diamond fly-cut V-groove in electroless nickel after polishing using a conical pin type tool (polishing time 5 min, polishing force 0.2 N, rotational speed of the polishing spindle 300 rpm, relative velocity 0.02 m/s). The roughness R_a of the fly-cut groove of about 14 nm was improved by a factor of two. Furthermore, one can see the high form accuracy at the bottom of the V-groove. Roughness values R_a of the polished steel substrates are in the same order using both pin type and wheel type tools.

Abrasive flow machining of structured surfaces

Abrasive flow machining is a process actually used for improving the surface quality of inner profiles as well as for deburring and rounding of edges [3]. An abrasive medium consisting of a polymer fluid, abrasives with a defined grain size and additives is

pressed along the contours at a certain pressure and temperature. For these purposes the workpiece is positioned in a special workpiece holder and clamped between the upper and the lower cylinder of the abrasive flow machine tool. Then, the heated abrasive medium is cyclically pressed from one cylinder into the other passing the surface of the workpiece at a defined speed. In abrasive flow machining the material removal rate is in direct proportion to the speed of the abrasive medium [3].

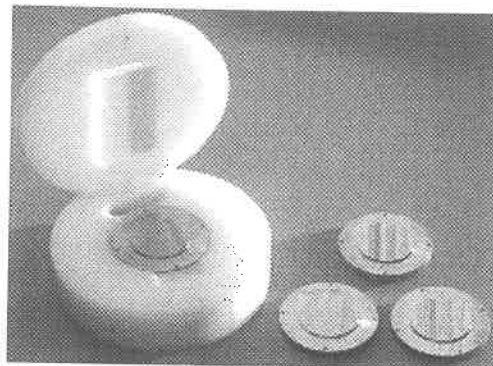


Fig. 4: Abrasive flow machined workpieces and tool holder.

For polishing of the structured steel and electroless nickel plated workpieces by abrasive flow machining the tool holder defining the pass of the fluid was made of polyamid (fig. 4). For machining the abrasive medium (50% SiC, 200 mesh) was heated up to 35°C. Machining time was 15 min for 5 cycles and the machining pressure was 1.5 MPa. Fig. 5 shows WLI-images of an abrasive flow machined V-groove in steel. As can be seen the edges at the bottom and top of the groove are rounded. Surface roughness Ra at the sides amounts to about 100 nm. Roughness Ra of the sides of the electroless nickel grooves is about 150 nm.

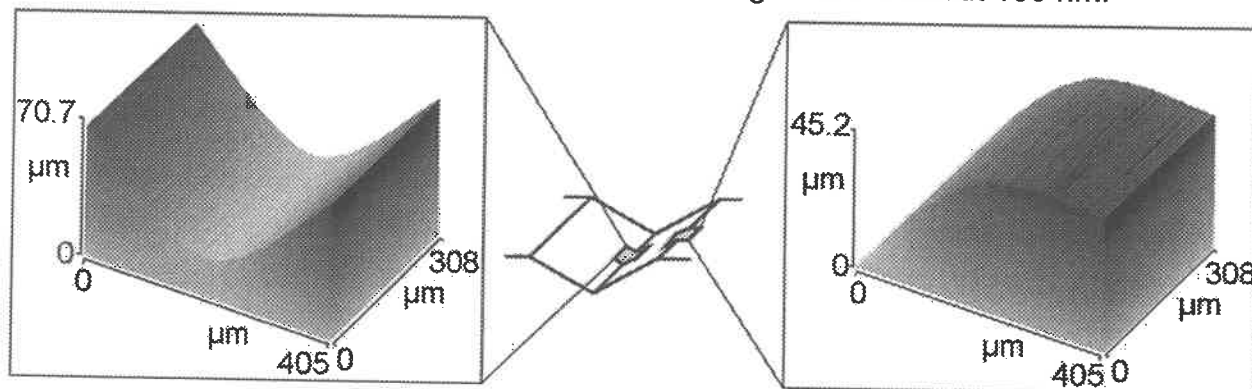


Fig. 5: Abrasive flow machined V-groove in steel, bottom (left) and top (right) of groove.

Laser polishing

Laser polishing of structured surfaces is performed by focussing a laser beam perpendicularly on the workpiece. In dependence of the power of the laser beam the surface of the workpiece is evaporated, i. e. material is removed, or slightly melted. In this case material is no longer removed and due to surface tensions of the melted material a smoothing effect can be achieved.

In first experiments a Nd:YAG-laser ($\lambda = 1064 \text{ nm}$) was used for laser polishing of a steel substrate with linear V-grooves (fig. 1). The feed velocity of the laser beam was varied in the range of 80 mm/min to 500 mm/min and the power of the laser beam between 50 W and 100 W. The diameter of the laser beam was 200 μm . Laser polishing was performed in three steps from "rough" polishing to finishing by reducing the laser power. Fig. 6 shows the laser polished steel sample (left) and a WLI-image of one side

of a 150° V-groove (right). The roughness Ra of the grooves is in the range of 300 nm to 450 nm. The melted and solidified surface can clearly be seen.

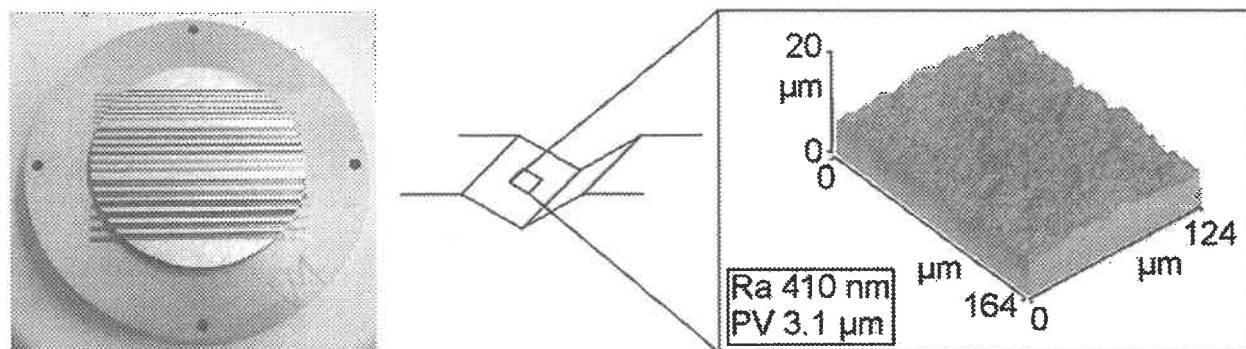


Fig. 6: Laser polished structured steel sample.

Conclusion and outlook

The new developed (chemo-) mechanical polishing process is well suited to improve surface roughness and to guarantee high form accuracy of the structured electroless nickel plated and steel substrates. The reduction of the structures' dimensions as well as polishing of circular, e. g. Fresnel structures are main objectives for future work. Due to its process technology abrasive flow machining is limited to more or less linear structures. Furthermore, the rounding of edges can not be avoided. Nevertheless, abrasive flow machining of structures will be continued using finer abrasives and optimized process parameters. Also, laser polishing experiments will be continued using optimized process parameters and to improve surface roughness and form accuracy. First results show the potential of laser polishing for finishing of structures but optical surface quality could not be achieved.

Acknowledgement

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PAD CONTACT AREA CHARACTERIZATION IN CHEMICAL MECHANICAL PLANARIZATION USING MICRO MOLDING TECHNOLOGY

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Introduction

Chemical Mechanical Planarization (CMP) is considered to be one of the most versatile IC fabrication technologies capable of achieving planar surfaces that are essential for very large scale integrated circuits (VLSI). During the CMP process, a wafer is placed face side down on a pad under high pressure, both rotating simultaneously. High pressure, rotational speed, and chemical-mechanical reactions enhanced by chemical slurry result in a planarized surface [1].

Generally, the pad surface has numerous independent pores that temporarily hold fresh slurry. The pores are separated by polyurethane wall structures that contain micro asperities. As these micro asperities provide space for reactions between wafer and pad, deterioration occurs continuously during the process [2]. The real contact area increases when the asperities make direct contact with the wafers and this results in reduced real contact pressure. As a result, the material removal rate drops [2]. To achieve constant removal rate and uniformity, it is required to maintain a constant real contact area. This is ensured by the use of a diamond grit conditioner that regenerates the peaks and valleys on the pad.

Since the real contact area plays an important role in CMP, pad surface characterization has been thought of as being very essential, but the pad surface deterioration mechanism has yet to be fully understood.

The deterioration of the pad surface in CMP is investigated in this paper using pads made with different geometries, which provide different deterioration mechanisms, wear and deformation. These design and fabrication of these pads (using micro molding technology) is described.

Pad deterioration mechanisms

Although the pad degrades during CMP, it is hard to establish an appropriate mechanism due to the inherent wet nature of the process. Wear and deformation are two candidate mechanisms explored in this paper and either one can theoretically explain the observed increase in real contact area. Until now, wear based on Greenwood & Williamson [3] and deformation on polymer erosion have been suggested [4]. The wear mechanism increases the number of contact points without changing the unit size of contact. Deformation, on the other hand, increases the contact area instead of the contact number. Figure 1 shows the two different pad deterioration mechanisms.

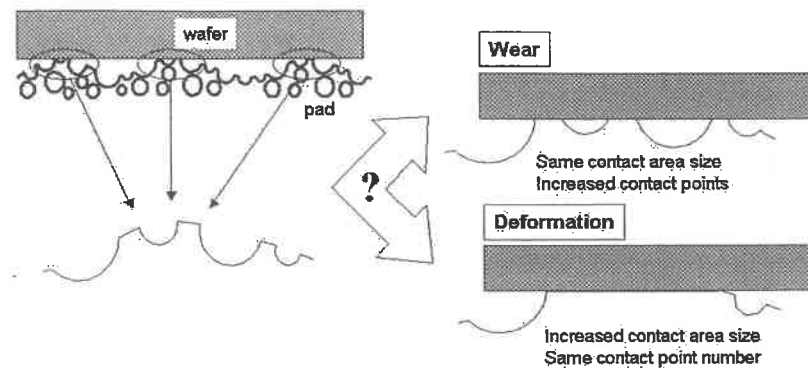


Figure 1. Pad deterioration mechanisms

Pad design based on a characterization

To understand the pad deterioration mechanism, it is necessary to characterize the pad surface topography. In this work, a VSI-based laser interferometer, WYKO NT3300, is used to measure pad surface topography. To improve reflectance, the pad is coated with 25nm gold layer. Generally a pad is made of polyurethane foam and has a surface of peaks and valleys after conditioning in an initial step.

From a top view analysis, the surface is covered with pores (diameter: 40-60 μm) to be separated with wall structures (width: 10-30 μm). The density of pores is known to be 30-50% (figure 2 (a)) Using surface profile analysis, a correlation between pad geometry and the conditioner is found. That is, there is a functional frequency in pad peaks and valleys. Based on this investigation, the side view of a pad can be divided into 3 regions, namely the reaction region, the transition region and the reservoir region (figure 2(b)). The reaction region is the space where pad and wafer have multiple direct contacts with abrasives. The transition region is for slurry transportation from the reservoir region to the reaction region. Fresh slurry supplied to the pad can be held temporarily in the reservoir region and flows through the transition region to the reaction region. As the degradation mechanism is limited to the reaction region, the discussion in this paper is restricted to that area. The transition and reservoir regions will be discussed later.

By using bearing ratio analysis, the data is acquired for the ratio of the area along the z-axis to the whole surface. Pad deformation along the z-axis is measured to be 10-12 μm when the pressure is 500g/cm². The ratio of the area 10 μm from the top to the whole surface is found to be 3-5% on average. A ratio of 3.2% is chosen for the initial state in pad design. The size of the contact area is measured to be 20-50 μm on each side. The distribution of each contact area is considered to be uniform. According to this analysis, the initial contact area is characterized as a series of units, each consisting of 5 individual squares (20x20 μm for each square) within a larger square region of 250x250 μm .

Two types of pads are designed and fabricated to simulate the two pad degradation mechanisms. The wear mechanism can be captured suitably by increasing the contact point numbers of the designed pad. Pads with increased contact area but identical contact point number can mimic the effects of the deformation mechanism. Experiments are performed with pads that simulate these two conditions. The ratio of real contact area to pad surface area is maintained when investigating the two different mechanisms to enable a direct comparison of results. Pads used in the experiment are fabricated using micro molding technology.

Figure 3 shows the design of the pad.

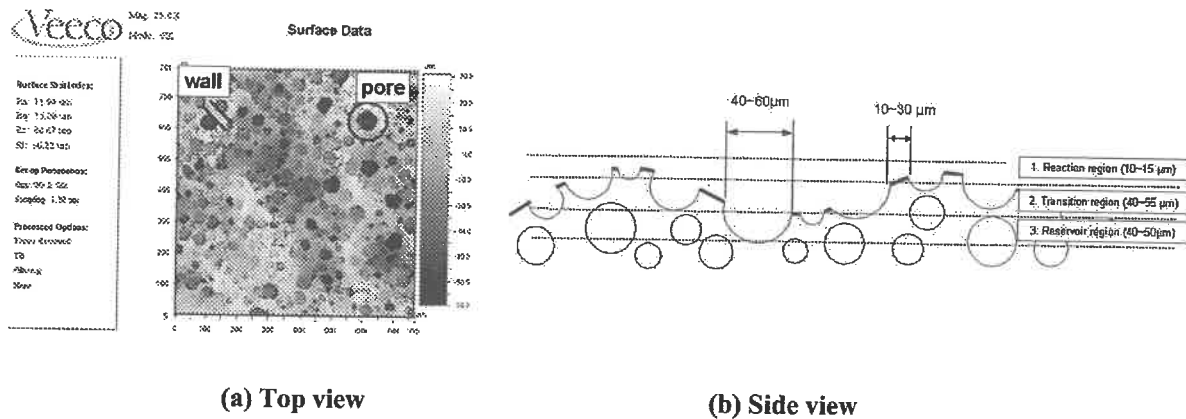


Figure 2. Pad surface analysis

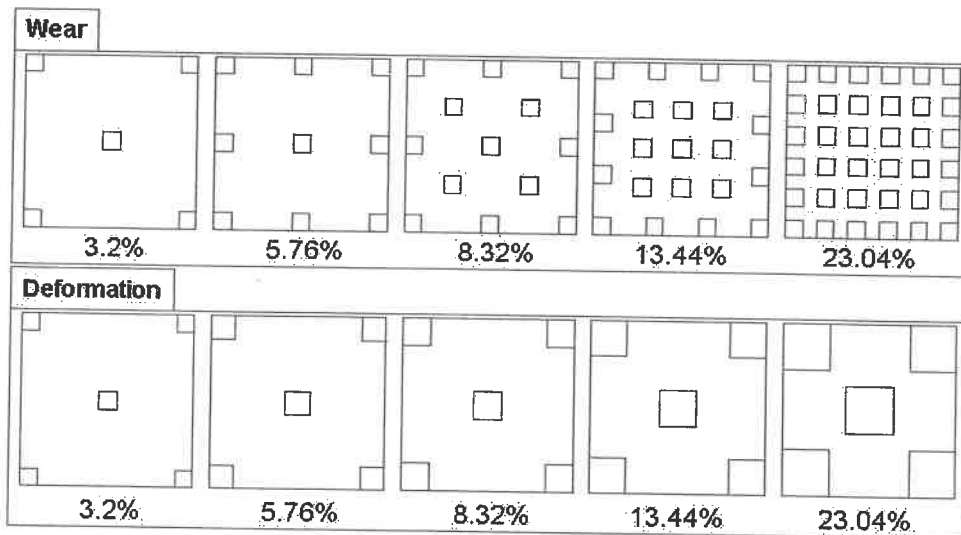


Figure 3. Pad designs

Pad fabrication

To verify wear and deformation theories, a micro molding technology is used in fabrication of the pads. A silicon wafer is etched with deep reactive ion etching (DRIE) process. The contact area is left unetched with a step height of $20\mu\text{m}$. This etched wafer is used as a master and replicated by silicone rubber in a molding process. Then the silicone rubber is used as the mold for the pad. Figure 4 details the indirect casting process.

After manufacturing the pad, the dimension of the etched wafer, silicone rubber, and pad is measured. In the etched wafer, the step height is $19.6\mu\text{m}$. This is precisely copied to a silicone rubber mold with a step height of $19.7\mu\text{m}$. Using this mold, the pad is fabricated with a step height of $19.1\mu\text{m}$. The Precision of the micro molding process is better than $1\mu\text{m}$. Figure 5 details the results of the measurement.

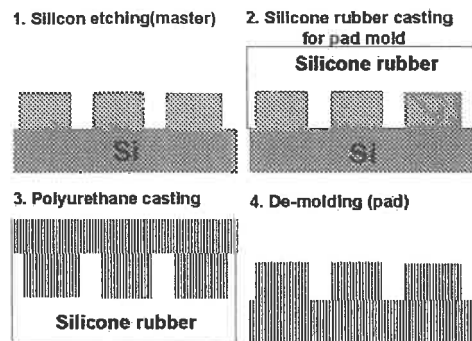


Figure 4. Pad fabrication process

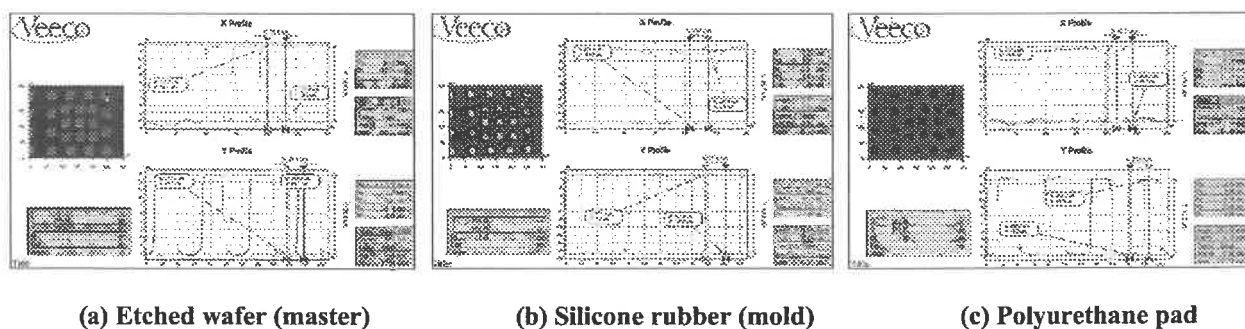


Figure 5. Dimension measurement

Conclusion & future work

To characterize the pad deterioration mechanism, pads with different geometries are fabricated using micro molding technology. The dimensional accuracy is far superior compared to other molding methods. CMP experiments are being planned, in order to clarify the pad degradation mechanism and will be reported on later.

Acknowledgements

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DEVELOPMENT OF CAVITATION AIDED MACHINING FOR FINISHING FINE SURFACE

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1. Introduction

Cavitation was discovered at the end of the 19th century. For more than one century since the discovery, mechanical engineers have been troubled with a cavitation which causes a generation of the erosion, vibration and noise of a fluid machinery.

In this paper, the cavitation aided machining as a precision manufacturing method is proposed, based on the new concept using positively the cavitation in a suction flow for machining phenomenon. That is to say, this manufacturing method makes loose abrasive grains, which are mixed with water as a carrier, to interfere in the workpiece surface with impact of the cavitation and flow of the machining fluid restricted partially by the restrictor above a machining point on workpiece surface. A fundamental machining phenomenon in cavitation aided machining of glass surface is made clear and its machining performance is experimentally investigated by analyzing surface finish, stock removal and cavitation impact. Then an efficiency of applying the cavitation aided machining as a precision surface finishing is proved by carrying out surface finishing on glass surface in the order of nanometers based on the machining phenomenon.

2. Principle of Cavitation Aided Machining and Experimental Procedure

Fig. 1 shows a schematic view of the fundamental apparatus of the cavitation aided machining. This apparatus mainly consists of a chamber of machining fluid, a restrictor and a suction pump. When machining fluid filled in the chamber is sucked with a pump, cavitation occurs partially in a suction flow under a restrictor setting in the passageway as shown in this figure. This machining method makes loose abrasive grains, which are mixed with machining fluid, to interfere the workpiece surface with impact of the cavitation and flow of the machining fluid by the restrictor just above workpiece surface. Main machining conditions are shown in table 1. The shape of restrictor is a semicircular disc of 20mm in radius. The suction pump used in test has an ability of 0.075m³/min in rated discharge and 7.5m in head-capacity. Workpiece surface is polished down to about 9.3nmRy in surface finish before tests. Surface finish is measured with Rank Taylor Hobson Nanostep. In this machining, the interference intensity of loose abrasives is influenced by the cavitation impact generated by collapsing bubbles. However, it is too difficult to measure the intensity of cavitation impact because it increases up to several giga-pascals during several micro-seconds in very small area¹⁾. The cavitation intensity is measured by a miniature quartz force transducer (Kistler Type 9211) fixed under a hole of 1.3mm in bore drilled through from workpiece surface. Measured data is stored in a digital storage oscilloscope (200MHz in sampling rate).

3. Machining Phenomenon and Performance in Cavitation Aided Machining

Fig. 2 shows the variation of stock removal S under a restrictor in cavitation aided machining. The stock removal is extremely small just under the restrictor tip ($y=0$) and it increases with increasing the distance from restrictor tip y within 12.5mm in y in which stock

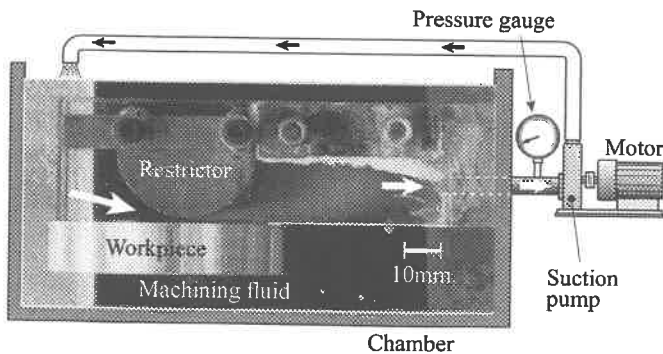


Fig. 1 Schematic view of the fundamental apparatus of the cavitation aided machining

Table 1 Main machining conditions

Abrasive grains	WA600, WA1000, WA2500, WA4000, WA8000, GC4000, AZ4000
Machining fluid	Water
Concentration of grain	30 %wt.
Workpiece	Soda-lime-glass (120 x 60 x 5 mm)
Clearance of restrictor	2 mm
Width of passageway	6 mm
Flow rate	10.6 l/min
Machining time	30 min

removal is maximum. In order to make clear such peculiar variation of stock removal, the distribution of cavitation impact P_c on the workpiece surface in a machining process is shown in this figure. The cavitation impact is slightly generated under a restrictor tip ($y=0$). It increases with increasing the distance from restrictor tip and has the maximum at about 12.5mm in y . Furthermore it decreases with increasing y in downstream form its maximum point. Such variation of cavitation impact is similar to that of stock removal and the point of maximum cavitation impact closely agree with that of maximum stock removal. This suggests as follows. Bubbles with cavitation are mainly generated and grow up near the restrictor tip, and then bubbles collapse after flowing away from the restrictor tip because of increasing pressure of machining fluid by increasing the cross-sectional area of passageway. At the moment of collapsing bubbles, the impact for interfering acts loose abrasive grains flowing in machining fluid.

Fig. 3 shows the effect of loose abrasive grain size on stock removal S at the workpiece surface downstream of the restrictor tip. Stock removal increases with larger grain size, and workpiece material just under the restrictor tip ($y=0$), on which cavitation impact weakly generates, is remarkably removed because of the flow of machining fluid mixed with abrasive grains in the cases of grain size larger than #2500. In any cases of grain size, the stock removal increases with increasing the distance from restrictor tip y and it has the maximum at about 12.5mm in y . Such variations of stock removal depend on the distribution of cavitation impact on machining area of workpiece surface as shown in fig. 2.

Fig. 4 shows the effect of loose abrasive grain size on surface finish R_y at the surface downstream of the restrictor tip. The initial surface roughness of 9.3nm in maximum height is shown with a straight dotted line. The surface finish decreases with finer grain size and its variation is not similar to the stock removal. Improving surface finish down to about 5nm in maximum height partially, cavitation aided machining with WA4000 has a most suitable effect for finishing glass surface. Surface finish with #8000, almost as large as the initial surface roughness, is larger than that with #4000. Such results suggest that the kinetic energy of each acting loose abrasive grain supplied by sucked cavitation flow decreases with decreasing the grain size. However, the finishing ability of cavitation aided machining with WA8000, which is about 1.2 μ m in average diameter, is lower than that with another grain size because of too small kinetic energy of abrasive grain for enough machining effect in 30 minutes.

Fig. 5 shows a DFM observation of finished glass surface with WA4000. A fine surface, with a lot of very small

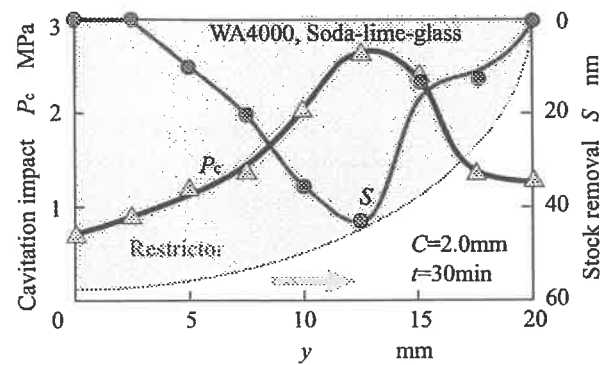


Fig. 2 Variations of stock removal or cavitation impact under a restrictor

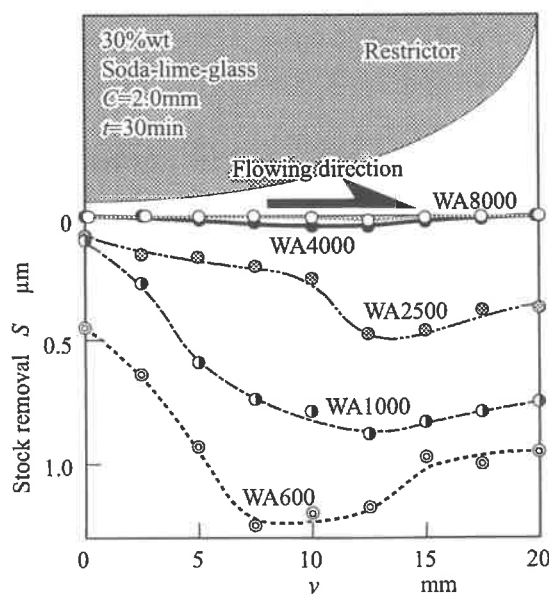


Fig. 3 Effect of loose abrasive grain size on stock removal

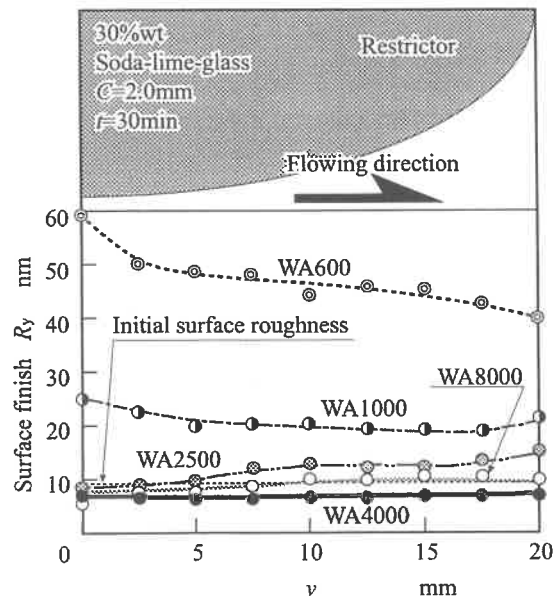


Fig. 4 Effect of loose abrasive grain size on surface finish

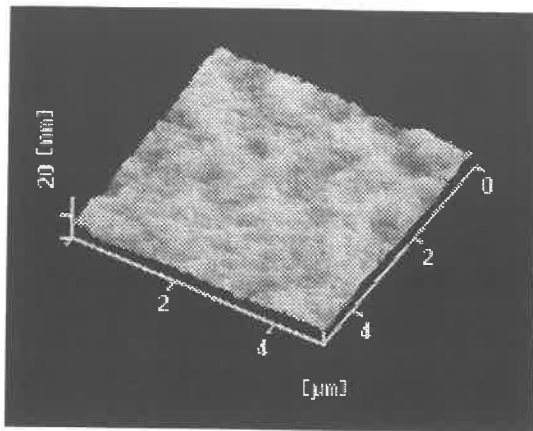


Fig. 5 DFM observation of machined glass surface with WA4000

Table 2 Characteristics of loose abrasive grain used in test

Grain	WA	GC	AZ
Specific gravity	3.98	3.98	4.22
Knoop hardness	2050	2550	—

dimples of several nano-meters in depth, is generated by cavitation aided machining. It is considered that such small dimples are generated by mild interference of loose abrasive grains applied a cavitation impact on workpiece surface.

Fig. 6 shows the effects of abrasive grain material on stock removal S and surface finish R_y . And **table 2** shows characteristics of three kinds of abrasive grain used in test. AZ grain is produced by mixing alumina abrasive with zirconia abrasive of 30%wt. which is about 1300 in knoop hardness²¹.

The stock removal becomes the maximum at 12.5mm in y with any kinds of abrasive grains because of the maximum cavitation impact at the point. And the stock removal with AZ grain is larger than that with WA grain which is similar to that with GC grain. Such results indicate that the kinetic energy of AZ grain applied with cavitation and flowing of machining fluid is larger than those of another grains because of its larger specific gravity. While the surface finish becomes smaller in the order, GC, AZ, WA grain, and surface finish with GC grain is especially larger than the initial surface roughness. It is considered that taking part each other, the effect of characteristics of abrasive grains appeared on such result of surface finish. That is, the interference intensity of GC grain is larger than of WA grain because of higher hardness of GC grain. AZ grain acts on the workpiece surface stronger than WA grain, as the result that AZ grain including zirconia abrasive is softer than WA grain but its specific gravity is larger than that of WA grain. Therefore cavitation aided machining leads the mechanical material removal in very small scale by interference of loose abrasive grains.

4. Finishing Fine Surface

The above mentioned investigation makes clear the fundamental machining phenomenon in cavitation aided machining of glass and its effective conditions for finishing fine surface. So fine glass surface should be finished by applying such machining phenomenon.

Fig. 7 shows a schematic view of the surface finishing apparatus by the cavitation aided machining. Main constructions of the apparatus are almost as same as shown in **Fig. 1** except for a restrictor slid parallel to the workpiece surface along the flowing direction in the extended intake by a ball screw and a synchronous motor. Main finishing conditions are shown in **table 3**.

Fig. 8 shows variations of stock removal S in surface finishing process by cavitation aided machining. The restrictor slides once back and forth between 70mm and 0mm in y with a setting rate v_r . The stock removal is approximately

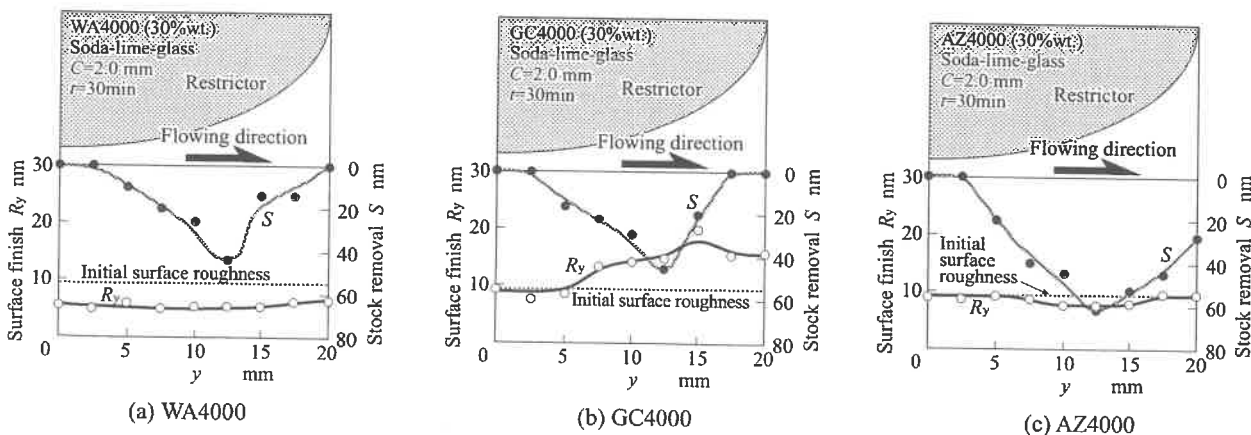


Fig. 6 Effects of loose abrasive grain on stock removal and surface finish

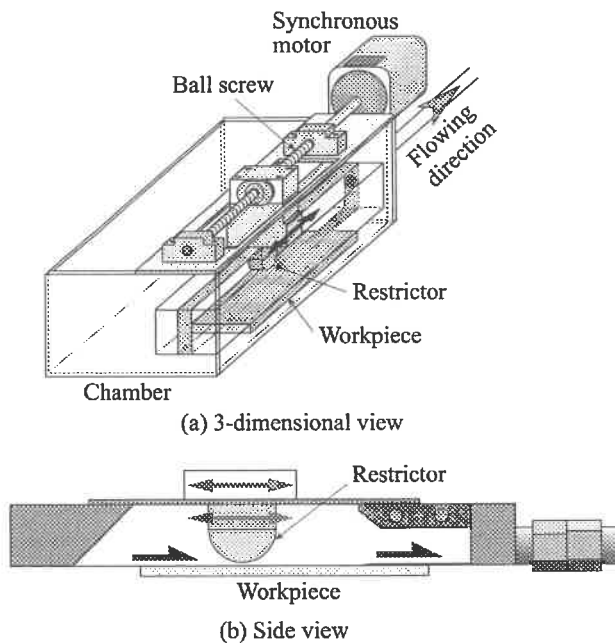


Fig. 7 Schematic view of the surface finishing apparatus by the cavitation aided machining

Table 3 Main finishing conditions

Abrasive grains	WA4000,
Workpiece	Soda-lime-glass (120 x 60 x 5 mm)
Sliding rate of restrictor	0.8, 1.2, 1.6 mm/min
Sliding distance	70mm x 2 passes

constant except near both sliding ends of restrictor at any sliding rates. In the case of 0.8mm/min in sliding rate, fine surface less than 20nm in flatness is generated through about 60mm in length.

Fig. 9 shows the effect of surface finish R_y on sliding rate of restrictor v_r in surface finish process by cavitation aided machining. Surface finish decreases with decreasing sliding rate of restrictor and is smaller than that of initial surface. In the case of 0.8mm/min in v_r , in which stock removal is largest in the setting region of sliding rate in the test, surface finish is improved down to 7.6nm in maximum height but is larger than that in the fundamental test shown in fig. 4. Therefore such results will indicate the possibility of ultraprecision finishing of glass surface by cavitation aided machining.

5. Conclusions

Main conclusions obtained in this paper are as follows.

- (1) Cavitation aided machining, as improved the surface finish of glass down to about 5.0nm in maximum height by using abrasives of WA 4000, has a possibility for applying to an ultraprecision finishing.
- (2) WA grains generate finer machined surface than GC abrasives that is harder or AZ abrasives that have higher specific gravity.
- (3) Fine finished glass surface with 60mm in length is generated by applying a fundamental machining phenomenon of cavitation aided machining along flowing direction of machining fluid continuously.

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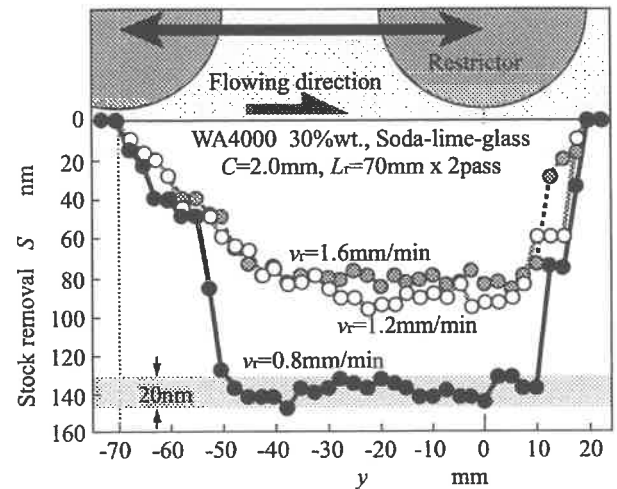


Fig. 8 Variations of stock removal in surface finishing process

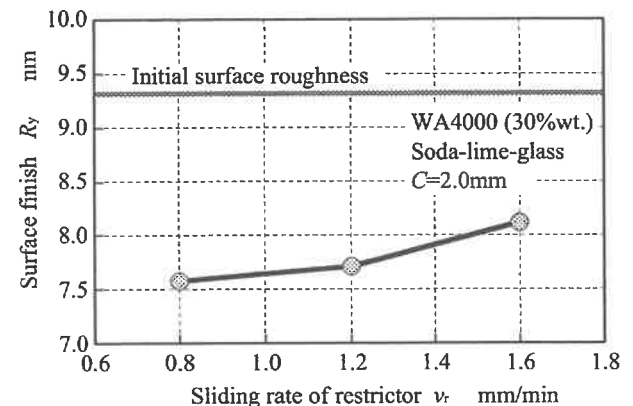


Fig. 9 Effects of surface finish on sliding rate of restrictor in surface finishing process

A New Lapping Film with Agglomerative Super Fine Abrasives for Large-Diameter Workpiece Surface Finishing

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Abstract

A lapping film with agglomerative superfine abrasives was previously reported on its excellent finishing properties: achieving scratch-free optical mirror surface quality at a much higher finishing rate compared with the traditional slurry/pad finishing. However, finishing properties over a long process time and the possibility in applying it to large-diameter workpiece surface finishing, were left unexamined. In this study, the finishing properties over a long finishing time are investigated. Different from the conventional fixed-abrasive tools, self-dressing of the agglomerative abrasives occurs during the process, allowing excellent surface roughness obtained continuously at a relative stable removal rate. And, it is found that film indexed Web-type finishing is effective to avoid the influence of the abrasive wear and maintain the initial high removal rate with the finishing time. Furthermore, the possibility to achieve excellent surface quality with high removal rate in large-diameter workpiece is also verified from the results of $\Phi 200\text{mm}$ P-TEOS film finishing.

Key words: Polishing, Surface finishing, Lapping film, Fixed-abrasive, Slurry, Agglomerative abrasive, Abrasive wear, Self-dressing

1. Introduction

In our previous study [1], a lapping film with agglomerative superfine abrasives was introduced for optical glass surface finishing. Compared with the traditional pad/slurry finishing, the lapping film exhibited results that the same excellent scratch-free optical mirror surface quality could be achieved at a much higher finishing rate. However, according to the conceptual model as illustrated in Figure 1, the removal rate would vary over time, since the contacting areas changes due to the abrasive wear during the process. This study focuses on examinations of the changes of the finishing properties over a longer finishing time together with the abrasive wear, and some approaches to make the removal rate maintain stable. Finally, in order to verify the possibility in

applying to large-diameter workpiece, surface finishing in $\Phi 200\text{ mm}$ P-TEOS films with this new lapping film is also performed and the results are reported.

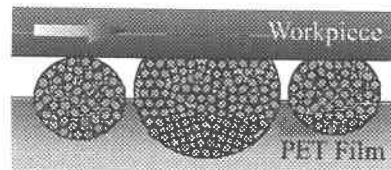
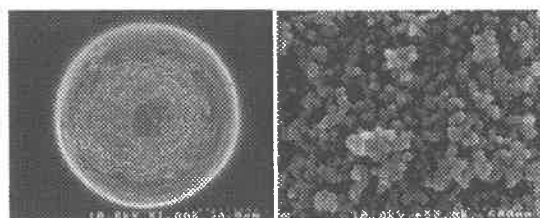


Fig.1 Conceptual model of finishing with the agglomerative abrasives

2. Experimental procedure

The abrasive grains (median diameter: $60\text{ }\mu\text{m}$) with no binding material inside were agglomerated from ZrO_2 superfine particles (Figure 2). As shown in Table 1, two types of lapping film with different

crushing resistance were prepared. The crushing resistance of the agglomerated grain (tensile strength in compression to fracture), tested on a Shimazu MCT500 (Micro Compression Testing Machine), was examined to evaluate the bonding force among the primary super fine particles.



A) Agglomerated grain B) Super fine particles

Fig. 2 SEM images of ZrO_2 agglomerated grain and super fine primary particles

Table 1 Specifications of the lapping films

Symbol	RZ-1	RZ-2
Crushing resistance of abrasive	20 MPa	60 MPa
Abrasive content	40-45area%	
Binding resin	Polyurethane	
Backing material	PET film (Thickness:75 μ m)	

Table 2 Basic finishing conditions

Workpiece	Optical glass (BK7) Φ 12 mm (3-piece chucked)
Finishing pressure	66 kPa
Plate (Film) rotation	60 rpm
Workpiece rotation	36 rpm
Finishing fluid	Pure water – 1mL/min

Finishing experiments on optical glasses were performed on a Lapmaster-type rotary lapping machine with the lapping films pasted onto the lapping plate. Table 2 shows the basic finishing conditions. The surface roughness before finishing of all the workpieces was adjusted to 30 nmRy. And for comparison, the traditional slurry/pad polishing with CeO_2 was also carried out under the same conditions.

3. Finishing properties over a long finishing time

As plotted in Figure 3, surface roughness of all the workpieces is maintained at the previous value after 20 minutes finishing. While, as to the removal rate, compared with the relatively constant values of the CeO_2 slurry/pad finishing, both two lapping films show a high rate initially, followed by a sharp falloff, and then decrease slightly after 5 minutes. And, RZ-2 shows a much higher removal rate, twofold at 2.5 minutes and approximately the same after 5 minutes, compared with the CeO_2 slurry/pad finishing.

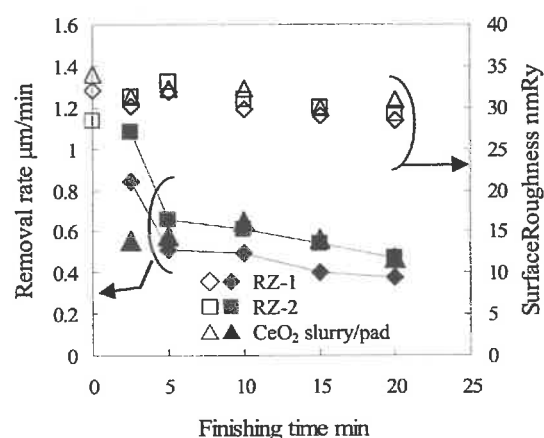


Fig.3 Changes of finishing properties with finishing time

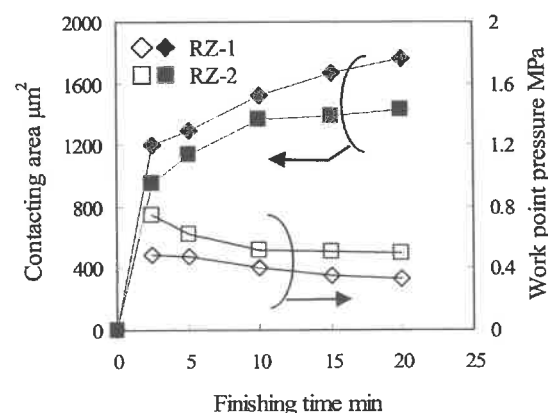
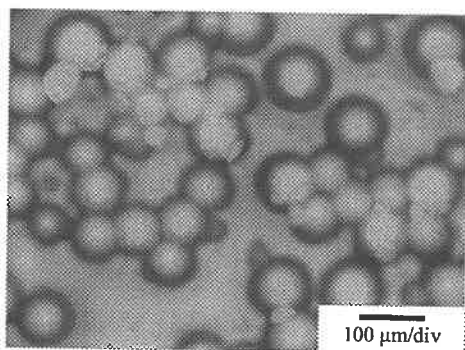


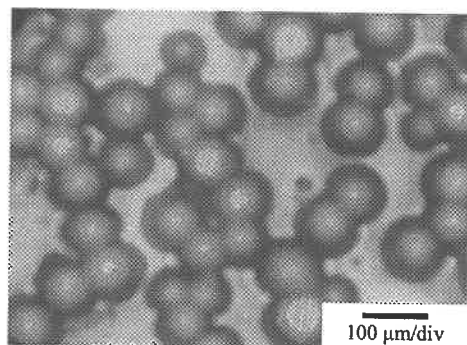
Fig.4 Changes of abrasive wear and working point pressure with finishing time

As shown in Figure 4, corresponding to the removal rate, the contacting area measured from the flat circle

area at the top of the spherical abrasive, increases drastically at the beginning, and then goes on slowly and continuously. In other words, the agglomerative abrasives wear quickly at first due to the high working point pressure, after the abrasive grains truncated flat, the contacting area increases and thus the pressure at the working points turns lowered so that the abrasive wear is suppressed. However, there is no absolute stop in abrasive wear and it still goes on slightly and continuously. Therefore, it is believed that such kind of self-dressing behavior allows new cutting edges to emerge to working points and make the pressure at the working points decrease slightly, resulting in a relatively constant removal rate to achieve excellent surface roughness stably and continuously.



A) Top view of RZ-1 film after 20 minutes finishing



B) Top view of RZ-2 film after 20 minutes finishing

Fig. 5 Abrasive wear after 20 minutes finishing

On the other hand, as explained previously [1], the difference in crushing resistance of the agglomerative

abrasive is considered as the reason for the difference in removal rate between RZ-1 and RZ-2. As shown in Figure 5, RZ-1 with weaker crushing resistance wears obviously much faster. This indicates that the RZ-1 was rather destroyed so that the contacting area was much larger and pressure at the working points turns lower, resulting in a lower removal rate.

4. Effect of film index on the removal rate

Although self-dressing could be achieved to obtain a relatively stable removal rate after the abrasive truncated, the initial high removal rate was considered difficult to maintain in this kind of rotary finishing. In order to achieve the initial high removal rate continuously, it is expected that a reel-to-reel WEB-type finishing would be effective. Experiment with 1-piece chucked $\Phi 50$ mm BK7 glass disks was carried out by using RZ-2 film under the conditions shown in Table 3. For comparison, experiment performed on Lapmaster-type lapping machine was also carried out under the same conditions listed in Table 2 except for the finishing pressure is set to 12 kPa.

As shown in Figure 6, in the case of film indexed, the removal rate maintains at an approximately constant value, namely $0.16 \mu\text{m}/\text{min}$. In contrast, in the case of rotary finishing, the removal rate declines with the finishing time just the same way as reported in chapter 3.

Table 3 Finishing conditions in Web-Type finishing

Finishing pressure	12 kPa
Workpiece rotation	30 rpm
Workpiece oscillation	100 mm/min
Film index rate	100 mm/min
Finishing fluid	Pure water – 1 mL/min

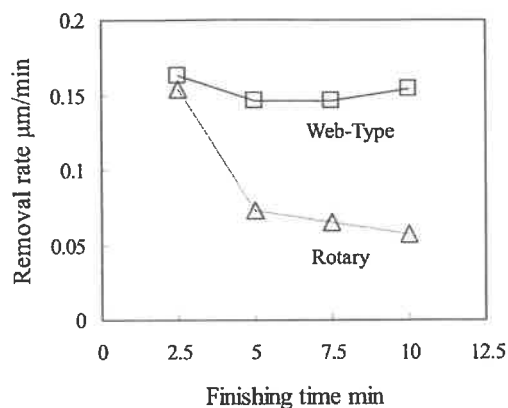


Fig.6 Effect of the film index on the removal rate

5. Finishing properties in $\Phi 200$ mm P-TEOS film

Finishing experiments on $\Phi 200$ mm P-TEOS films were performed on an ARW-681MS CMP machine (MAT inc., JAPAN). Finishing pressure was set to 30kPa and only pure water was used as finishing fluid. For comparison, slurry/pad finishing with SS25 silica slurry (Cabot Microelectronics Corp.) and IC1000/SUBA400 (Rodel Corp.) was also carried out.

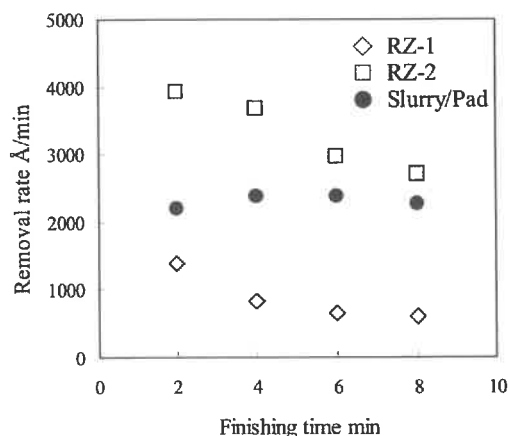


Fig. 7 Changes of the removal rate with the finishing time in P-TEOS film finishing

Figure 7 exhibits that the removal rate of both lapping films behaves in the same way as that obtained in the BK7 optical glass, declining with the finishing time. And RZ-2 shows much higher rate than that in

the slurry/pad finishing. It could be expected that this decline in removal rate would be possibly improved by using film indexed web-type method as reported in chapter 4. As to the uniformity across the wafer, about 3% is obtained, which is as the same as that obtained by slurry/pad finishing. Furthermore, the surface roughness studied by using a Zygo Newview5032, is about 8 nmPV and 6 nmPV in an observed 0.36*0.27mm area, respectively for RZ-2 and slurry/pad process after 10 minutes finishing.

6. Conclusions

Different from the conventional fixed-abrasive process, self-dressing effect could be achieved in agglomerative abrasives without conditioning required. It is also found that film indexed Web-type finishing is an effective way to avoid the influence of the abrasive wear and maintain the initial high removal rate over long finishing time. Although further improvements are required, the possibility of this new lapping film to obtain excellent surface quality with higher removal rate in $\Phi 200$ mm P-TEOS film finishing is also clarified. There are obvious follow-up tests are already underway such as running more wafers under constant conditions, and defect levels, killer and micro scratches, are also being pursued.

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